TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS.

VOL. XXVI.

LTH MEETING, NEW YORK, N. Y., 1904.
LIST MEETING, SCRANTON, PA., 1905.

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OFFICERS

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

1904–1905,

FORMING THE STATUTORY COUNCIL.

PRESIDENT.

JOHN R. FREEMAN ..... Providence, R. I.

VICE-PRESIDENTS.

D. S. JACOBUS ..... Hoboken, N. J.
M. L. HOLMAN ..... St. Louis, Mo.
WILLIAM J. KEEP ..... Detroit, Mich.

Terms expire at Annual Meeting of 1905.

S. M. VAUCLAIN ..... Philadelphia, Pa.
H. H. WESTINGHOUSE ..... Pittsburg, Pa.
FRED. W. TAYLOR ..... Philadelphia, Pa.

Terms expire at Annual Meeting of 1906.

MANAGERS.

R. C. MCKINNEY ..... New York, N. Y.
S. S. WEBBER ..... Trenton, N. J.
NEWELL SANDERS ..... Chattanooga, Tenn.

Terms expire at Annual Meeting of 1905.

GEORGE I. ROCKWOOD ..... Worcester, Mass.
JOHN W. LIEB, JR. ..... New York, N. Y.
ASA M. MATTICE ..... Pittsburg, Pa.

Terms expire at Annual Meeting of 1906.

GEO. M. BRILL ..... Chicago, Ill.
FRED. J. MILLER ..... New York, N. Y.
RICHARD H. RICE ..... Lynn, Mass.

Terms expire at Annual Meeting of 1907.

TREASURER.

WM. H. WILEY ..... Nos. 43–45 East 19th St., New York, N. Y.

SECRETARY.

PROF. F. R. HUTTON ..... No. 12 West 31st St., New York, N. Y.

HONORARY COUNCILLORS.

PAST PRESIDENTS OF THE SOCIETY.

THURSTON, R. H.	1880—1882	Died Oct. 25, 1903
LEAVITT, E. D.	1882—1883	Cambridge, Mass.
SWEET, JOHN E.	1883—1884	Syracuse, N. Y.
HOLLOWAY, J. F.	1884—1885	Died Sept. 1, 1896
SELLERS, COLEMAN	1885—1886	Philadelphia, Pa.
BABCOCK, GEORGE H.	1886—1887	Died Dec. 16, 1893
SEE, HORACE	1887—1888	
TOWNE, HENRY R.	1888—1889	New York, N. Y.
SMITH, OBERLIN	1889—1890	Bridgeton, N. J.
HUNT, ROBERT W.	1890—1891	Chicago, Ill.
LORING, CHARLES H.	1891—1892	Brooklyn, N. Y.
COXE, ECKLEY B.	1892—1894	Died May 13, 1895
DAVIS, E. F. C.	1894	Died Aug. 6, 1895
BILLINGS, CHARLES E.*	1895	Hartford, Conn.
FRITZ, JOHN	1895—1896	Bethlehem, Pa.
WARNER, WORCESTER R.	1896—1897	Cleveland, O.
HUNT, CHARLES WALLACE	1897—1898	New York, N. Y.
MELVILLE, GEORGE W.	1898—1899	Philadelphia, Pa.
MORGAN, CHARLES H.	1899—1900	Worcester, Mass.
WELLMAN, S. T.	1900—1901	Cleveland, O.
REYNOLDS, EDWIN	1901—1902	Milwaukee, Wis.
DODGE, JAMES M.	1902—1903	Philadelphia, Pa.
SWASEY, AMBROSE	1903—1904	Cleveland, O.

[Note.—According to the Constitution, Article C 27, the five surviving Past Presidents who last held the office shall be members of the Council, with all the rights, privileges and duties of the other members of the Council.]

* Unexpired term of E. F. C. Davis.

NOTE.

The considerable bulk of the volume of Transactions has induced the Publication Committee to direct the insertion of a summary of the Society membership in place of the complete list of members which was published in the earlier volumes. The summary attaching to this issue is that which appears in the catalogue of the Society issued with corrections to July, 1905. Reference for complete list should be made to the "Geographical List" for July, 1905.

Foreign Countries.

	Membership.		Membership.
Africa	19	Holland	1
Australia	6	India	2
Belgium	4	Jamaica, W. I.	1
Canada	31	Japan	5
Central America	1	Mexico	8
China	1	Norway	1
Cuba	4	Russia	5
France	8	South America	7
Germany	10	Sweden	3
Great Britain (England)	45	Switzerland	1
Great Britain (Scotland)	4	Trinidad, B. W. I.	1

Total Foreign Membership.....168

United States.

	Membership.		Membership
Alabama	8	Nebraska	4
Alaska	1	Nevada	1
Arizona	2	New Hampshire	16
Arkansas	3	New Jersey	146
California	39	New York	827
Colorado	23	North Carolina	3
Connecticut	125	North Dakota	1
Delaware	12	Ohio	206
District Columbia	27	Oklahoma	2
Georgia	13	Oregon	5
Hawaiian Islands	1	Pennsylvania	385
Illinois	184	Porto Rico	1
Indiana	34	Rhode Island	51
Iowa	5	South Carolina	3
Kansas	6	South Dakota	1
Kentucky	3	Tennessee	7
Louisiana	16	Texas	5
Maine	15	Utah	2
Maryland	36	Vermont	13
Massachusetts	256	Virginia	27
Michigan	74	Washington	7
Minnesota	12	West Virginia	10
Missouri	43	Wisconsin	70
Montana	11		

Total Membership in the United States 2,742

GEOGRAPHICAL SUMMARY.

Total Foreign Membership	168
Total Membership in United States	2,742
*Present Address Unknown	5
Total Membership	2,915

SUMMARY OF MEMBERSHIP BY GRADES.

Honorary Members	20
Members	1,985
Associates	259
Junior Members	651
Total Membership	2,915
†Life Members	109

* These are Chas. J. Hillard, Chas. R. Johnston, John O. Norbom, Anthony Victorin, Members, and L. G. C. Mayer, Junior, and if any member knows their present addresses he will confer a favor by so advising the Secretary.

† These Life Members are included in the total membership above, in the class to which they belong.

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AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

CONSTITUTION.

NAME, OBJECT AND GOVERNMENT.

C 1. The title of this Society is "The American Society of Mechanical Engineers."

C 2. The object of the Society is to promote the Arts and Sciences connected with Engineering and Mechanical Construction. The principal means for this purpose shall be the holding of meetings for the reading and discussion of professional papers, and for social intercourse; the publication and distribution of its papers and discussions; and the maintenance of an Engineering Library.

C 3. The Society shall be governed by this Constitution, and by By-Laws and Rules in harmony therewith.

C 4. The Society was organized as a Corporation under the laws of the State of New York, April 7, 1880. Its offices shall be located in the City of New York.

MEMBERSHIP.

C 5. Persons connected with the Arts and Sciences relating to Engineering or Mechanical Construction may be eligible for admission into the Society.

C 6. The membership of the Society shall consist of Honorary Members, Members, Associates and Juniors. Honorary Members, Members and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.

C 7. Honorary Members, Members and Associates are en-

titled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member. A proxy shall not be valid for a greater time than six months.

C 8. Honorary Members shall be persons of acknowledged professional eminence, and their number shall not exceed twenty-five at any time.

C 9. A Member must have been so connected with Engineering as to be competent, as a designer or as a constructor, to take responsible charge of work in his branch of Engineering, or he must have served as a teacher of Engineering for more than five years. A Member shall be thirty years of age or over.

C 10. An Associate must either have the other qualifications of a Member or be so connected with Engineering as to be competent to take charge of engineering work, or to co-operate with Engineers. An Associate shall be twenty-six years of age or over.

C 11. A Junior must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school. A Junior shall be twenty-one years of age or over.

C 12. The rights and privileges of every Honorary Member, Member, Associate and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

ADMISSION.

C 13. Honorary Members shall be nominated by at least ten members of the Society. The grounds upon which the nomination is made, shall be presented to the Council in writing.

C 14. All applications for membership to the grades of Member, Associate or Junior shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council, shall, unless objection is made by the applicant, be submitted to the voting membership for election, by means of a letter-ballot.

C 15. Associates or Juniors desiring to change their grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C 16. Election to membership shall be by a sealed letter-ballot as the By-Laws shall provide. Adverse votes to the number of two per cent. of the votes cast shall be required to defeat the election of an applicant for any grade of membership. The Council, may in its discretion, order a second ballot upon a defeated applicant, in which case adverse votes to the number of four per cent. of the votes cast, shall be required to defeat the election.

C 17. The election of Honorary Members shall be by a vote of the Council taken by letter-ballot, as provided in the By-Laws. One dissenting vote shall defeat such election.

C 18. Each person elected, excepting Honorary Members, shall subscribe to this Constitution, and shall pay the initiation fee before he can be entitled to the rights and privileges of membership. If such person does not comply with this requirement within six months after notice of his election, he will be deemed to have declined election. The Council may, thereupon, declare his election void.

INITIATION FEES AND DUES.

C 19. The initiation fee for membership in each grade shall be as follows:

For Member.....Twenty-five Dollars,
For Associate.....Twenty-five Dollars,
For Junior.....Fifteen Dollars.

C 20. A Junior, on promotion to any other grade of membership, shall pay an additional fee of Ten Dollars.

C 21. The annual dues for membership in each grade shall be as follows:

For Member.....Fifteen Dollars,
For Associate.....Fifteen Dollars,
For Junior.....Ten Dollars for the first six years of his membership and thereafter the same as for an Associate.

C 22. The Council may in its discretion, permit any Member or Associate to become a Life Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the

annual dues in his grade. Such Life Member shall not be liable thereafter for annual dues.

C 23. The Council shall have the power, by letter-ballot, to admit to Life Membership, without the payment of a life membership fee, any person who, for a long term of years, has been a Member or an Associate when, for special reasons, such procedure would, in its judgment, promote the best interests of the Society, provided that notice of such proposed action shall have been given at a previous meeting of the Council. One dissenting vote shall defeat such admission.

SUSPENSIONS AND EXPULSIONS.

C 24. Any Member, Associate or Junior who shall leave his annual dues unpaid for one year, shall not receive the volume of *Transactions* until such arrears are paid. Any Member, Associate or Junior who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.

C 25. The Council may refuse to receive the dues of any member of any grade, who shall have been adjudged by the Council to have violated the Constitution or By-Laws of the Society, or who, in the opinion of the Council by a two-thirds vote, shall have been guilty of conduct rendering him unfit to continue in its membership; and the Council may expel such person and remove his name from the list of members.

THE COUNCIL.

C 26. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer and five Past Presidents. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the Council and take part in the discussion of financial questions but shall not have a vote.

C 27. The five surviving Past Presidents who last held the office shall be members of the Council with all the rights, privileges and duties of the other members of the Council.

C 28. The Council thus constituted shall be the legal Trustee of the Society. All gifts or bequests not designated for a specific purpose shall be invested by the Council, and only the income therefrom may be used for current expenses.

C 29. Should a vacancy occur in the Council, or in any elective office except the presidency, through death, resignation or other cause, the Council may elect a Member or Associate to fill the vacancy until the next annual election.

C 30. The Council shall regulate its own proceedings, and may by resolution delegate specific powers to an Executive Committee or to any one or more members of the Council. No act of the Executive Committee or of a delegate shall be binding until it has been approved by a resolution of the Council.

C 31. The Council shall present at the Annual Meeting of the Society a report verified by the President or Treasurer or by a majority of the members of the Council, showing the whole amount of real and personal property owned by the Society, where located, and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated or expended during the year immediately preceding such date, and the purposes, objects or persons to or for which such applications, appropriations or expenditures have been made; also the names and places of residence of the persons who have been admitted to membership in the Society during the last year, which report shall be filed with the records of the Society, and an abstract thereof shall be entered in the minutes of the proceedings of the Annual Meeting.

C 32. An act of the Council, which shall have received the expressed or the implied sanction of the membership at the next subsequent meeting of the Society, shall be deemed to be the act of the Society, and shall not afterwards be impeached by any member.

C 33. The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member or Associate to fill the vacancy

until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

OFFICERS.

C 34. At each Annual Meeting there shall be elected from among the Members and Associates:

A President to hold office for one year.

Three Vice-Presidents, each to hold office for two years.

Three Managers, each to hold office for three years.

A Treasurer to hold office for one year.

C 35. The election of officers shall be by sealed letter-ballot, as the By-Laws shall provide.

C 36. The term of all elective officers shall begin on the adjournment of the Annual Meeting of the Society. Officers shall continue in their respective offices until their successors have been elected and have accepted their offices.

C 37. A President, Vice-President or Manager shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

C 38. The Council, at its first meeting after the Annual Meeting of the Society, shall appoint a person of the grade of Member to serve as Secretary of the Society for one year, subject to removal for cause by an affirmative vote of fifteen members of the Council, at any time after one month's written notice has been given him to show cause why he should not be removed, and he has been heard in his own defense, if he so desires. The Secretary shall receive a salary which shall be fixed by the Council at the time of his appointment.

C 39. The President, Secretary and Treasurer shall perform the duties legally or customarily attaching to their respective offices under the Laws of the State of New York, and such other duties as may be required of them by the Council.

C 40. A vacancy in the office of President shall be filled by the Vice-President, who is senior by age.

MEETINGS.

C 41. The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at

such time and place as the Council may appoint. Fifty Members and Associates shall constitute a quorum for the transaction of business.

C 42. Special meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty members entitled to vote, the notices for such meetings to state the business for which such meeting is called, and no other business shall be entertained or transacted at that meeting.

C 43. Any appropriation recommended by the Society at a meeting shall not take effect until it has been approved by the Council.

C 44. Every question which shall come before a meeting of the Society or of the Council or a Committee, shall be decided by a majority of the votes cast, unless otherwise provided in this Constitution or the By-Laws, or the Laws of the State of New York. The Council may order the submission of any question to the membership for discussion by letter-ballot. Any meeting of the Society at which a quorum is present, may order the submission of any question to the membership for discussion by letter-ballot.

STANDING COMMITTEES.

C 45. The Standing Committees of the Society to be appointed by the President shall be:

Finance Committee,
Committee on Meetings,
Publication Committee,
Membership Committee,
Library Committee,
House Committee.

C 46. There shall be a John Fritz Medal Committee of three members appointed as provided in the By-Laws.

C 47. The Annual Committees shall be:

An Executive Committee, appointed by the Council.
A Nominating Committee, appointed by the President.
Tellers as required by the By-Laws, appointed by the President.

C 48. Special Nominating Committee:

Twenty or more members entitled to vote may constitute

themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.

C 49. Professional Committees:

The Council shall have power to appoint, upon a recommendation of the Society at a general meeting, or upon its own initiative, such Professional Committees as it may deem desirable, to investigate, consider and report upon subjects of engineering interest. Reports of such committees may be accepted by the Society and printed in the *Transactions*, but shall not be approved or adopted as the action of the Society. Any proposed expenses of such committees must be authorized by the Council before they are incurred.

C 50. Each Committee shall perform the duties required of it in the By-Laws, or assigned to it by the Council. The Secretary of the Society shall be the Secretary of each of the Standing Committees.

C 51. The Council may at any time, in its own discretion, remove any or all members of any Committee, except a Nominating Committee; and the vacancy, arising from this or from any other cause, shall be filled by appointment by the President, except a vacancy in the Executive Committee, which shall be filled by the Council.

SECTIONS OF THE SOCIETY.

C 52. The Council may, in its discretion, authorize the organization of sections or groups of any or all grades of membership, for professional or scientific purposes which are in harmony with the Constitution and By-Laws of this Society. Such sections or groups may, in the discretion of the Council, be geographical or professional, and shall have such powers, and act under such rules and regulations as the Council may from time to time prescribe.

TRANSACTIONS.

C 53. The official record of technical papers and discussion, shall be known as the *Transactions* of the Society, and shall be published under the direction of the Council. There may be included therein, the annual report of the Council, reports of Committees, and business records of the Society.

C 54. The Society shall claim no exclusive copyright to any papers read at its meetings, or any reports or discussions thereon, except in the matter of their official publication under the Society's imprint as its *Transactions*. The policy of the Society shall be to give the professional and scientific papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging Engineering progress and extending the professional reputation of its members.

C 55. The Society shall not be responsible for statements or opinions advanced in papers or in discussions at its meetings. Matters relating to politics or purely to trade shall not be discussed at a meeting of the Society, nor be included in the *Transactions*.

C 56. The Society shall not approve or adopt any standard or formula, or approve any engineering or commercial enterprise. It shall not allow its imprint or name to be used in any commercial work or business.

AMENDMENTS TO THE CONSTITUTION.

C 57. At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted on at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at twelve o'clock noon on the first Monday of March following.

C 58. The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the

votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

AMENDMENTS TO BY-LAWS AND RULES.

C 59. For the further ordering of the affairs of the Society, the Council may, by a two-third vote of its members present, amend the By-Laws in harmony with this Constitution, provided that a written notice of such proposed amendment shall have been given at the previous regular meeting of the Council; and provided further that the Secretary shall have mailed to each member of the Council a copy of such proposed amendment, at least thirty days in advance of the meeting of the Council at which action is to be taken. The amendment shall take effect immediately on its passage by the Council. The Secretary shall at once mail a copy of such amendment to the members of all grades.

C 60. The Council may, by a majority vote of the members present at any meeting, establish, amend or annul Rules for the conduct of the business affairs of the Society; for the ordering and conduct of its professional or business meetings; and for guidance of its committees in their work and reports; provided that such Rules are in harmony with the Constitution and By-Laws of the Society.

CONSTITUTION GOES INTO EFFECT.

C 61. This Constitution shall supersede all previous Rules of the Society, and shall go into effect on the announcement by the Presiding Officer of its adoption.

BY-LAWS.

CANDIDATES FOR MEMBERSHIP.

B 1. A candidate for admission to the Society as a Member or as an Associate must make application on a form approved by the Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Con-

stitution, By-Laws and Rules of the Society. He must refer to at least five Members or Associates to whom he is personally known.

B 2. Applications for membership from Engineers who are not resident in the United States or Canada, and who may be so situated as not to be personally known to five Members of the Society, as required in the foregoing paragraph, may be recommended for ballot by five members of the Council, after sufficient evidence has been secured to show that in their opinion the applicant is worthy of admission to the grade which he seeks.

B 3. A candidate for admission to the Society as a Junior must make application in the same manner as provided for Members, except that he must refer to not less than three Members or Associates to whom he is personally known.

B 4. References shall not be required of candidates for Honorary Membership.

B 5. The references for each candidate for admission to the Society shall be requested to make a confidential communication to the Membership Committee, setting forth in detail such information, personally known to referee, as shall enable the Council to arrive at a proper estimate of the eligibility of the candidate for admission to the Society.

ELECTION OF MEMBERS.

B 6. The Secretary shall mail to each member entitled to vote, at least thirty days in advance of each annual or semi-annual meeting, a ballot stating the names and the respective grades of the candidates for membership in the Society which have been approved by the Council, and the time of the closure of voting. The voter shall prepare his ballot by crossing out the names of candidates rejected by him, and shall enclose said ballot in a sealed blank ballot envelope which he shall then enclose in a second sealed outer envelope on which he shall, for identification, write his name in ink. The ballot thus prepared and enclosed shall be mailed or delivered unopened to the Tellers of Election. The Secretary shall certify to the competency, and the signature of all voters. On the closure of voting, the Tellers of Election shall first open and destroy the outer envelopes, and shall then canvass the ballots, and certify the result to the meeting of the Society.

B 7. The Tellers shall not receive any ballot after the stated time of the closure of voting. A ballot without the endorsement of the voter, written in ink on the outer envelope, is defective, and shall be rejected by the Tellers of Election.

B 8. The names of those persons elected to membership, with their respective grades, shall be embodied in a written report, signed by the Tellers, and presented to the next meeting of the Society. The President shall then declare them duly elected to membership in the Society. The Tellers may, through the Secretary, in advance of any meeting advise each candidate of the result of the canvass of the votes in his case. The names of applicants who are not elected shall neither be announced nor recorded in the *Transactions*.

B 9. The endorsers of an applicant who has not been elected, may, with his consent, present to the Council a written request for a re-submission of his name to ballot. The Council may, in its discretion, by a three-fourths vote of the members present, order the name of the applicant placed on the next ballot for members.

B 10. Election to Honorary Membership shall be by letter-ballot of the Council. A notice of such proposed election shall be mailed by the Secretary to each member of the Council at least sixty days in advance of the date set for the closure of such election.

B 11. Each person elected to membership, except an Honorary Member, must subscribe to the Constitution, By-Laws and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society.

ELECTION OF OFFICERS.

B 12. The Secretary shall mail to each member entitled to vote, at least thirty days before the Annual Meeting, the names of the candidates for office proposed for election by the Nominating Committee.

B 13. The names of the candidates proposed by the Nominating Committee or Committees, and the respective offices for which they are candidates, shall be printed in separate lists on the same ballot sheet, each list of candidates to be printed under the names of the members of the particular committee which proposed it.

B 14. The name of any candidate on the ballot may be erased, and the name of any person qualified to hold the office written in its stead. The ballot thus prepared must be voted and canvassed in the same manner as for the election of members.

B 15. At the first session of the Annual Meeting, the Tellers of Election of Officers shall canvass the votes cast for the officers of the Society in the manner prescribed for the election of members, and immediately report the result of the canvass to the meeting. The President shall then announce the candidates having the greatest number of votes for their respective offices, and declare them elected for the ensuing year.

B 16. In case of a tie in the vote for any officer, the President or, in his absence, the Presiding Officer shall cast the deciding vote.

B 17. A ballot which contains more names marked by a cross on it than there are officers to be elected, is thereby defective, and shall be rejected by the Tellers.

FEES AND DUES.

B 18. The initiation fee and annual dues of the first year shall be due and payable on notice of election to membership, and upon that payment the member will be entitled to the *Transactions* for the year. Thereafter the annual dues shall be due and payable on the first day of October in each year.

B 19. A member in arrears for one year shall not be entitled to vote until such arrears have been paid. Should the right to vote be questioned, the books of the Society shall be conclusive evidence.

B 20. The Secretary shall present to the Council the name of any Member, Associate or Junior in arrears for more than one year, and such member shall not receive the *Transactions* until such arrears are fully paid. A person dropped from the rolls for non-payment of dues may, in the discretion of the Council, be restored to the privileges of membership, upon payment of all arrears.

FINANCIAL ADMINISTRATION.

B 21. The Council at its first meeting in each fiscal year, shall consider the recommendations of the Finance Committee

concerning the expenditure necessary for the work of the Society during that year. The apportioning of the work of the Society among the various Standing and other Committees shall be on a basis approved by the Council and in harmony with the Constitution and By-Laws. The appropriations approved by the Council, or so much thereof as may be required for the work of the Society, shall be expended by the various Committees of the Society, and all bills against the Society for such expenditure shall be certified by the Committee making the expenditure and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employee of the Society except upon bills duly audited by the Finance Committee, or by resolution of the Council.

COMMITTEES.

B 22. The President within one month after the Annual Meeting shall fill all vacancies in the Standing Committees by appointment from the membership of the Society.

Each of the Standing and the Annual Committees, shall, at their first meeting after the Annual Meeting, elect a Chairman to serve for one year. The President shall appoint the Chairman of each Professional Committee. A member of a Standing Committee whose term of office has expired, shall continue to serve until his successor shall have been appointed.

FINANCE COMMITTEE.

B 23. The Finance Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. This Committee shall, in the discretion of the Council, have a supervision of the financial affairs of the Society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent public accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for presentation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate of the prob-

able income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and, when called upon by the Council, advise upon financial questions.

COMMITTEE ON MEETINGS.

B 24. The Committee on Meetings shall consist of five persons who may be members of any grade. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Committee to procure professional papers, to pass upon their suitability for presentation, and to suggest topical subjects for discussion at the meetings. The Committee may refer any paper presented to the Society to a person or persons, especially qualified by theoretical knowledge or practical experience, for their suggestions or opinions as to the suitability of the paper for presentation. Papers from non-members shall not be accepted except by unanimous vote of the Committee.

The Committee shall arrange the programme of each meeting of the Society, and shall have general charge of the entertainments to be provided for the members and guests at each meeting. It shall prohibit the distribution or exhibition at the headquarters or at the meeting places of the Society of all advertising circulars, pamphlets or samples of commercial apparatus or machinery. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

PUBLICATION COMMITTEE.

B 25. The Publication Committee shall consist of five Members or Associates. The term of office of one member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers or discussions, or parts of the same, shall be printed in the *Transactions* of the Society. The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

MEMBERSHIP COMMITTEE.

B 26. The Membership Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of this Committee:

To meet monthly to receive and scrutinize all applications for membership to the Society.

To send to each voting member the name, qualifications, engineering experience and references of each applicant, together with extracts from the Constitution and By-Laws relating to membership.

To seek further information as to the qualifications of an applicant, whose evidence of eligibility is not clear to the Committee.

To report to each session of the Council the names of all applicants under consideration together with the action of the Committee on each.

The Committee shall at once destroy all correspondence in relation to each applicant when his name has been placed on the ballot by order of the Council, or upon the withdrawal of the application.

LIBRARY COMMITTEE.

B 27. The Library Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Library Committee to take charge of the Library of the Society, the historical relics, the paintings and objects of art, and to recommend to the Council suitable regulations for their care and use. At the end of each fiscal year, the Committee shall deliver to the Secretary, a detailed report of its work.

HOUSE COMMITTEE.

B 28. The House Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the House Committee to have the care, management and maintenance of the house of the Society and its furnishings. They may make rules for the care and the use

of the Society House, subject to the approval of the Council. At the end of each fiscal year, the Committee shall deliver to the Secretary a detailed report of its work.

EXECUTIVE COMMITTEE.

B 29. The Council shall appoint from its members an Executive Committee to act for the Council during the interval between its sessions. The Committee shall make a report of its acts to each session of the Council for approval. The Secretary may take part in the deliberations of the Executive Committee, but shall not have a vote therein.

NOMINATING COMMITTEES.

B 30. A Nominating Committee of five Members, not members of the Council, shall be appointed by the President within three months after he assumes office. It shall be the duty of this Committee to send to the Secretary on or before October first the names of consenting nominees for the elective offices next falling vacant under the Constitution. Upon the request of any Member or Associate, the Secretary shall furnish to the applicant the names of such nominees.

B 31. A special Nominating Committee if organized, shall, on or before October twentieth, present to the Secretary the names of the candidates nominated by it for the elective offices next falling vacant under the Constitution, together with the written consent of each.

JOHN FRITZ MEDAL COMMITTEE.

B 32. The John Fritz Medal Committee shall consist of three persons of the grade of Member, to be appointed by the Council. The term of office of one member of this Committee shall expire at the end of each annual meeting. The duty of this Committee shall be to represent the Society in the Board of Trustees of the John Fritz Medal Fund Corporation.

REPRESENTATIVE DELEGATES.

B 33. The Council may in its discretion appoint a member or members of the Society or other person or persons to repre-

sent it at meetings of Societies of kindred aim or at public functions. Such delegates shall be designated as "Honorary Vice-Presidents," and their duties shall terminate with the occasion for which they were appointed.

TELLERS.

B 34. The Presiding Officer shall, at the first session of the Annual Meeting, appoint three Tellers of Election of officers, whose duties shall be to canvass the votes cast, and report the result to the meeting. Their term of office shall terminate when their report of the canvass is presented to the meeting.

B 35. The President within one month after assuming office shall appoint three Tellers of Election of members to serve for one year, whose duties shall be to canvass the votes cast for members during the year, and to certify the same to the President. They shall notify candidates through the Secretary of the result of such election.

B 36. The President shall appoint three Tellers to canvass any letter-ballots which shall be ordered by the Council or by the Society.

MEETINGS.

B 37. The meetings of the Society shall continue from day to day as the meeting may decide. The business session of the Annual Meeting shall be held on Wednesday following the first Tuesday of December. The professional sessions for the reading of papers shall be held at such times and places as the meeting may appoint. Notices of all meetings of the Society shall be mailed by the Secretary to members of all grades not less than thirty days before the date of such meeting.

SECRETARY.

B 38. The Secretary of the Society shall be the Secretary to the Council and also to each of the Standing Committees.

The Secretary shall, under the supervision of the Finance Committee, have charge of the Books of Account of the Society.

He shall make and collect all bills against members or others.

He shall have charge of all bills against the Society, shall

keep an account of the same, and shall present them in proper form to the Finance Committee for audit.

All funds received by any person for the Society, shall be delivered to the Secretary. He shall immediately enter them in the Books of Account, and shall immediately deposit such funds as he receives, to the credit of the Society, in a Bank to be designated by the Council.

TREASURER.

B 39. The Treasurer shall make payments only on the audit of the Finance Committee, or upon the direction of the Council, by resolution of that body. He shall furnish a bond for the faithful performance of his duties to such amount as the Council may require, such bond to be procured from an incorporated Guarantee Company, at the expense of the Society.

TITLES, EMBLEMS, CERTIFCATE.

B 40. Each Member and Associate shall, subject to such rules as the Council may establish, be entitled on request, to a certificate of membership, signed by the President and Secretary of the Society. Every such certificate shall remain the property of the Society, and shall be returned to it on demand of the Council.

B 41. Each proxy authorizing a person to vote for an absent member, shall be signed by such absent member, with an attesting witness, and be submitted to the Secretary for verification of the member's right to vote at the meeting at which the right is to be exercised.

B 42. The emblem of each grade of membership approved by the Council shall be worn by those only who belong to that grade. The official stationary shall be used only by Officers and Committees of the Society.

B 43. The abbreviation of the titles of the various grades of membership approved by the Society are as follows:

For Honorary Members, . .	Hon. Mem. Am. Soc. M. E.
For Members, ...	Mem. Am. Soc. M. E.
For Associates, ...	Assoc. Am. Soc. M. E.
For Juniors, ...	Jun. Am. Soc. M. E.

RULES.

R 1. The Secretary's office shall be open on business days from 9 A.M. to 5.30 P.M. During the Annual Meeting, the office shall be open from 9 A.M. to 10 P.M. A register shall be kept for each regular meeting, to record the attendance of members and guests.

R 2. The Secretary shall provide a numbered badge or pin for each member or guest attending the regular meetings, the number on the badges to correspond with the member's of guest's number on the register.

R 3. The Secretary shall at each regular meeting of the Society distribute at the headquarters a printed list of the names registered at the meeting.

R 4. Copies of papers to be read and discussed at any meeting shall be sent to each member thirty days in advance of that meeting. A paper received too late for such distribution shall only be accepted for presentation at that meeting by unanimous consent of the Committee on Meetings. A blank shall accompany the papers by which a member may signify his intention to discuss any of the papers, and priority in debate shall be given in the order of the receipt by the Secretary of such notification.

R 5. At professional sessions, each paper shall be read by abstract only, ten minutes being allowed to the author for the presentation, unless otherwise ordered by the meeting.

R 6. A member who has given notice of his intention to discuss a paper, and shall have reduced his discussion to writing, shall be entitled to ten minutes for its presentation.

R 7. Each speaker shall be limited to five minutes in the oral discussion of a paper, unless the time should be extended by unanimous consent. A member who has once had the floor cannot claim it again until all the others have been heard who desire to speak on that paper. Authors may have five minutes to close the discussion on the paper.

R 8. Members unable to attend the meeting may send a discussion of any paper in writing, to be presented by the Secretary.

R 9. The Committee on Meetings shall deliver to the Secretary such papers as they recommend for presentation to the professional meetings of the Society.

R 10. The Secretary shall have sole possession of papers and illustrations between the time of their approval by the Committee on Meetings, and their presentation to the professional session of the Society.

R 11. After the presentation and discussion of a paper, a copy of both shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to each member who presented it, with the request that he correct errors or omissions, and return the same promptly to the Secretary.

R 12. Members may order reprints of papers at a price sufficient to cover the cost to the Society, provided that said copies are not for sale.

R 13. The Secretary may furnish to the author twenty copies of his paper without charge. He may also furnish to the technical press such papers in advance of the meeting as they may wish to publish after presentation to the meeting of Society.

R. 14. The entertainments to be provided for the members and guests at any meeting of this Society in any city shall be in charge of a Local Committee, subject, however, to the general approval of the Committee on Meetings.

R 15. A member may invite a non-member to the professional sessions of the meeting, but the guest shall not take part in the proceedings without an invitation from the Presiding Officer. Invitations to guests of members for the entertainments provided for the Society shall be in the discretion of the Local Committee.

R 16. The Society House shall be open at all hours for access to members. The Library shall be open on all week days between the hours of 10 o'clock A.M. and 10 o'clock P.M. It shall be conducted as a Free Public Reference Library of Engineering and the Allied Arts and Sciences.

R 17. Juniors who were elected to membership in the Society six years or more previous to the adoption of this Constitution, shall pay the same dues as an Associate, beginning with the fiscal year which opens after such adoption. Juniors, who have been elected less than six years before that date, shall pay the dues of an Associate on the expiration of six years after their election.

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PAPERS

OF THE

NEW YORK MEETING

(Lth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

DECEMBER 6th TO DECEMBER 9th, 1904

BEING ALSO THE TWENTY-SIXTH ANNUAL MEETING OF THE SOCIETY.

No. 1051.

PROCEEDINGS

OF THE

NEW YORK MEETING

(Lth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

December 6th to 9th, 1904.

The twenty-fifth annual meeting of the Society was held in the city of New York, pursuant to the provisions of the Constitution, beginning on the first Tuesday of December, 1904. The increasing size of the Society and its attendant increase in the numbers attending its annual meeting, made it necessary to provide for the holding of the large meetings in a hall outside of the Society House. The selection fell upon the hall of the Mendelssohn Union, at 113 West Fortieth Street, for the session on Wednesday and Thursday. The opening session on Tuesday evening and the closing session on Friday morning were held in the auditorium of the Society House, at No. 12 West Thirty-first Street, but were uncomfortably crowded at both sessions. Mr. Ambrose Swasey, President of the Society, presided at all the sessions.

At the opening session on Tuesday evening in the Society House, the members met for registration at headquarters and for social enjoyment until, at nine o'clock, they were called to order in the auditorium for the reading of the President's address. This was entitled "Some Refinements of Mechanical Science," and appears as one of the papers of the meeting.

The President at the close of his address called upon Past-President Charles Wallace Hunt, who presented to the Society for the donor a framed oil portrait of Past-President John E. Sweet. It will be remembered that Professor Sweet was one of those who, with Messrs. H. R. Worthington and Alex. L. Holley, were active in the first steps which resulted in the formation of the Society. It was made specially fitting by this circumstance that Mr. Hunt should have referred in his address to Past-President Sweet as one of the founders of the Society and as giving special significance to the presence of his portrait upon the walls of the convention room. Mr. Hunt's address of presentation is published as a paper of the meeting. President Swasey, in a few feeling words, accepted the gift as the official representative of the Society, and expressed his hope that for many years, as the members convened at the successful annual meetings of the Society, that Professor Sweet might be able to meet with them as he was present on this evening. After announcements by the Secretary, a recess was taken for luncheon in the supper room below.

Second Session. Wednesday Morning December 7th, 10 O'clock.

The second session of the annual meeting is by appointment of the constitution the business session of the meeting; pursuant to this arrangement, there was presented for record the annual report of the Council and its standing committees, which had been distributed in pamphlet form to all members in advance of the meeting. The annual report of the Council and of the committees are as follows:

ANNUAL REPORT OF THE COUNCIL.

Under the provisions of the Constitution the Council presents herewith a report of business transacted during the year, together with the various reports concerning the approved expenditure and the admissions to membership during the fiscal year ending September 30, 1904.

The amount of real and personal property owned by the Society, the property acquired during the year, the amount applied, appro-

priated or expended and the purposes and objects for which such expenditures have been made, will be found in the Report of the Finance Committee duly certified by the Audit Company of New York.

The Council appends to this report the names and residences of those who have been admitted up to and including September 30th, and submit these reports as abstracted, to be entered in the minutes of the Proceedings of the Annual Meeting.

The Council has held seven meetings during the year. The constitution of the body was altered by the provisions of the Constitution adopted at the annual meeting in December, 1903, and its first meeting was held under the provisions of the new constitution.

Messrs. James M. Dodge, D. S. Jacobus and John W. Lieb, with the President and Secretary, were appointed an Executive Committee. Mr. James M. Dodge was appointed representative of the Society on the John Fritz Medal Committee, to take the place of Professor Gaetano Lanza whose term was about to expire.

Messrs. James M. Dodge, Chas. Wallace Hunt and F. R. Hutton were confirmed as representatives of the Society on the Committee on the Joint Engineering Building.

Messrs. Swasey, Towne and Suplee have been appointed members of a Conference Committee to sit with appointees of the other two engineering societies who are to be founder societies in the Engineering Building, for the purpose of discussing and making recommendations as to the conduct of the Library which will result when the libraries of the founders' societies are brought together in one place.

Messrs. Swasey and Freeman were appointed delegates to represent the Society at the incorporation of Professor C. S. Howe, as President of the Case School of Applied Science in Cleveland, as Honorary Vice-Presidents under the provisions of the Constitution.

Messrs. Robert W. Hunt, Edwin Reynolds and Warren S. Johnson were appointed Honorary Vice-Presidents to represent the Society at the Fifty Year Jubilee of the University of Wisconsin.

Messrs. Woolson and Freeman were appointed delegates to the meeting of the National Fire Protection Association. Messrs. Charles E. Lucke, H. G. Chatain and Percy C. Idell were appointed tellers of election of members. As a Committee on Headquarters at the St. Louis Exposition, Messrs. Robt. H. Fernald, M. L. Holman and Wm. H. Bryan were appointed. Professor D.

S. Jacobus was appointed representative of the Society on a Joint Committee of the four Engineering Societies, on the subject of Standard Abbreviations.

Under the direction of the Council through the President of the Society as a special committee, four reunions of the members of the Society resident within convenient distance of New York City, were held on the last Tuesdays of the months of January, February, March and April. The speakers on these evenings who made the principal addresses were Messrs. John A. Brashear, whose address was entitled "The Evolution of Measurements"; Major Rogers Birnie, "Modern Ordnance"; Julian Kennedy, "How Steel Rails are Made"; and W. F. M. Goss, "The Modern Locomotive." The Council directed that the annual list of members which had hitherto been designated the catalogue of the Society, should hereafter be known as its "Year Book" and directed that the portraits of the officers of any Society year should be included in the matter published in its Year Book. The Council directed also that after the spring meeting a second edition should be issued in the form of a "vest pocket" or more compact geographical list, without the Constitution and By-Laws and without material concerning the exchanges in the Library.

The Council took up under the provisions of the Constitution the allocation of the different directions of expenditure to the various committees provided for in the Constitution.

It was decided that each standing committee should be responsible under the Council's general oversight for the expenditure appropriate to its particular work, and that such expenditure as did not fall to any other of the standing committees should be allocated to the Finance Committee for control. The Council confirmed also the system of requisitions and uniform vouchers reported by its Finance Committee.

The Council was requested by Mr. Andrew Carnegie, donor of the proposed building for the engineering societies to consider again the question of uniting with the Institute of Electrical Engineers, and the Institute of Mining Engineers, in the responsibility attaching to his gift of a building for the joint use of the engineering societies, in order that he might be definitely advised when the decision had been reached by the American Society of Civil Engineers, not to participate in the gift, that such decision did not invalidate the action taken by the Council in March, 1903.

The Council convened for this special purpose and passed the following minute:

"At a meeting of the Council of the American Society of Mechanical Engineers held in New York City, May 7th, 1903, the following action was taken:"

Resolved, That the American Society of Mechanical Engineers shall unite with the American Society of Civil Engineers, the American Institute of Electrical Engineers, the American Institute of Mining Engineers, and The Engineers' Club, or any of them, for the purpose of accepting the gift by Mr. Andrew Carnegie for the purpose of erecting suitable buildings for occupancy by various societies of engineers.

At a meeting of the Society held in Saratoga, New York, June 24, 1903, the following action was taken:

The American Society of Mechanical Engineers assembled in general session at its 47th meeting in Saratoga, N. Y., has learned with the greatest interest of the proposed gift to the profession of engineering by Mr. Andrew Carnegie, member of the Society, of a million dollars for an engineering building.

The Society has also been informed of the action taken by its Council in reference to making this gift available and serviceable to the needs of this Society. Wherefore:

Be it *Resolved*, that this Society desires to place on record its appreciation of the purpose of Mr. Carnegie, in seeking to advance by this means the interests of the profession of engineering,

Resolved, That by embodying this purpose in the form of a great and noble building for the uses of those organizations whose aims are to foster the development of engineering, the donor has taken a step which will notably advance these interests:

Resolved, That the Society approves the prompt response of its Council to the opportunity offered to favor and further the interests of the Society which are involved in that progress of the profession which lies at the base of the Carnegie gift.

Resolved, That it be referred to the Council with power to transmit by cablegram and letter to Mr. Carnegie the action of the Society, and to carry out by further action, the details necessary to realize Mr. Carnegie's generous purpose.

On motion the Society expressed its sentiment upon the gift, and the resolutions accepting it, by a rising vote, which was unanimous.

At a meeting of the Council held in New York on March 11, 1904, subsequent to publication of the decision of the American Society of Civil Engineers not to participate in the benefits of the proposed gift, the following action was taken:

Resolved, That the President be instructed to send a copy of the action of the Council on May 7th, and of the Society on June 24th, 1903, as contained in the foregoing minute, and certify thereto by affixing his signature and the seal of the Society.

Resolved, That the Council of the Society hereby reaffirms the resolutions passed May 7th, and June 24th, 1903, which accept in connection with The American Institute of Mining Engineers, and the American Institute of Electrical Engineers, the trust for engineering created by Mr. Carnegie.

Resolved, That the President be authorized, over his signature as President of the Society, to assure Mr. Carnegie of the earnest desire and unquestionable ability of this Society to accept and carry out all obligations incident to such acceptance and arising from it.

By order of the Council,

(Signed) AMBROSE SWASEY, *President*,

From the Minutes,

(Signed) F. R. HUTTON, *Secretary*.

The Council appointed a committee of residents of the city of St. Louis to have charge of the maintenance in Machinery Hall of the Louisiana Purchase Exposition of an office which should be a headquarters of the Society for the use and convenience of members during their visit to the exposition. The Council appropriated $500 for the expenses of such headquarters and referred the details to its Committee with power. The Committee with the co-operation of Mr. T. M. Moore, Chief of the Department of Machinery of the Exposition, arranged for the occupancy of a space in the gallery of Machinery Hall and completed arrangements with Mr. Kurt Toensfeldt to act as custodian of the headquarters, and provision was made by the interested courtesy of certain technical journals to have their publications on file in these headquarters. As the Exposition has not closed at the date of the preparation of this report, it is not possible to present any summary concerning the use which has been made of these provisions.

The Engineers' Club of St. Louis was invited by the Council to joint occupancy of its headquarters in Machinery Hall, if such arrangement should prove practicable and convenient.

The Society of Civil Engineers of France announced their intention to visit the United States in a somewhat organized party, to arrive on August 27th, and to remain until October 5th, following an itinerary laid out for them in advance. The Council directed that the courtesy of the facilities at the headquarters of the Society should be tendered to the French Society during their sojourn and that such arrangements as were in progress for the entertainment of the English engineers should also be put at the service of the visitors from France.

It will be recalled that the Chicago Meeting of the Society in June, 1904, was a joint meeting or the American Society and of

the British Institution of Mechanical Engineers. For the furthering of the purposes of the visitors from Great Britain, arrangements were made to organize headquarters in each important industrial city to which the visiting members might be assigned, by letters of introduction which should accredit them to the expert in the particular line of interest of the visitor in that city.

Such headquarters were appointed in the following centers:

CALIFORNIA:
Los Angeles—Henry E. Brett, 114 No. Spring Street.

COLORADO:
Denver—Frank E. Shepard, Denver Eng'g Works, 605 McPhee Bldg. cor. 17th and Glenwood Streets.

CONNECTICUT:
Hartford—Henry Souther, 440 Capitol Avenue.
Meriden—(See Hartford).
New Britain—(See Hartford).
New Haven—(See Hartford).
Norwich—(See Hartford).
Waterbury—(See Hartford).

DISTRICT OF COLUMBIA:
Washington—H. S. Knight, Room 313 McGill Bldg., 908 G. St., N. W.

ILLINOIS:
Chicago—R. W. Hunt, The Rookery and Auditorium Hotel, May 31st to June 3rd.
Urbana—L. P. Breckenridge, University of Illinois.

INDIANA:
Fort Wayne—H. J. Horstman, Bass Foundry & Machine Co.
Lafayette—W. F. M. Goss, Purdue University.

LOUISIANA:
New Orleans—W. B. Gregory.

MARYLAND:
Baltimore—B. J. Dashiell, Jr., Detrick & Harvey Machine Co., 508 E. Preston Street.

MASSACHUSETTS:
Boston—Jno. T. Boyd, care Yale & Towne Mfg. Co., 12 Pearl Street.
Cambridge—(See Boston).
Worcester—Headquarters Worcester Board of Trade. (Chas. E. Squier)

MICHIGAN:

DETROIT—Headquarters Board of Commerce, State Savings Bank Bldg (R. Ritchie).

MISSOURI:

ST. LOUIS—Engineer's Club, 709 Pine Street, and Machinery Hall, Exposition Grounds.

NEW JERSEY:

TRENTON—S. S. Webber, Trenton Iron Co.

NEW YORK:

BATAVIA—Chas. A. Alexander, Johnston Harvester Co.
BUFFALO—C. H. Bierbaum, 330 Prudential Bldg.
ELMIRA—N. B. Payne, The Payne Co.
NIAGARA FALLS—R. W. Emerson, Natural Food Co.
ROCHESTER—R. B. Cochrane, Cochrane-Bly Machine Works, 7 Griffith Street.
SCHENECTADY—H. G. Reist, General Electric Co.
SYRACUSE—Geo. H. Shepard, Syracuse University.
UTICA—Correl Humphrey, Chamber of Commerce.
WATERTOWN—L. P. Streeter, N. Y. Air Brake Co.

OHIO:

CINCINNATI—Headquarters Office of Thos. G. Smith, 412 Carlisle Bldg., 4th and Walnut Streets.
CLEVELAND—Ambrose Swasey, Warner & Swasey, Technical Club Rooms, Cleveland Arcade, 6th floor.
MANSFIELD—Jas. M. Brown, Office of Aultman & Taylor Machinery Co.
MASSILLON—Chas. O. Heggem, The Russell & Co.
SANDUSKY—W. H. Milspaugh, Sandusky Foundry & Machine Co.
TOLEDO—H. T. Yaryan, Toledo Heating & Lighting Co.
WELSTON—L. R. Shallenberger, Alma Cement Co.

PENNSYLVANIA:

ALTOONA—C. B. Dudley, Penn. R. R. Co.
DRIFTON—J. H. Pennington, Coxe Bros. & Co.
ERIE—J. R Wittemore, Erie City Iron Works.
FRANKLIN—B. Haskell, Franklin Rolling Mill & Foundry Co.
HARRISBURG—W. R. Fleming, Harrisburg Foundry & Machine Co.
JOHNSTOWN—Jos. Morgan, Jr., Cambria Steel Co.
OIL CITY—A. B. Steen, National Transit Co.
PHILADELPHIA—Engineers' Club.
PITTSBURGH—Westinghouse Electric & Manufacturing Co., Room 500, Westinghouse Building, Penn Avenue and Ninth Street.
SCRANTON—Alfred E. Lister, Room 31, D. & H. Depot or N. C. Durand.
So. BETHLEHEM—J. J. McKee, McKee & Milson.
WAYNESBORO—L. C. Nordmeyer, Frick Co.

RHODE ISLAND:
PROVIDENCE—W. A. Viall, Brown & Sharpe Mfg. Co.

WISCONSIN:
MADISON—W. B. & D. C. Jackson and Storm Bull.
MILWAUKEE—Geo. P. Dravo, 815 Herman Building.

CANADA:
MONTREAL—A. W. Robinson, 14 Phillips Sq., and Canadian Society Civil Engineers, 877 Dorchester Street.

In addition the Secretary's office provided a list of practitioners, firms or corporations in various lines of importance in the United States and these lists were given to the visitors according to his preferences or particular interests.

The Council are pleased to report that by the co-operation of the members in the various cities this plan of working to secure the purposes of the visits of these gentlemen seemed to be most acceptable and useful.

The authorities of the exposition at St. Louis had requested the American Society of Civil Engineers to nominate a group of representive members of the profession who would serve as a Committee on Organization for the holding of an Engineering Congress during the continuance of the Exposition period.

The chairman of this committee, Col. H. S. Haines, member of this Society, requested the Council to be represented at a meeting of the representatives of the four national societies, and certain other bodies, for a conference on the share which each constituent would be willing to assume of responsibility for such a congress. It was the desire of the organizing committee to repeat in 1904 the Chicago Engineering Congress of 1893. When the representatives were assembled it appeared that the Institute of Electrical Engineers had already completed arrangements for the holding of a congress in Electrical Engineering which was to have a distinctly international character. The Institute of Mining Engineers, and the Society of Mechanical Engineers were already under arrangements to conduct joint meetings with English or continental societies which were expecting to come to the United States during the summer in which the congress would naturally be held.

In view of these facts, and the Council not being advised that there had come before it any expressions from the profession at large, indicating its desire for the holding of such a Congress, or its interest therein, and in the absence of any provision from the

exposition authorities for the expenses unavoidably connected with the holding of such a Congress, the Council directed that the organizing committee be advised that this Society did not see its way clear to undertake to be responsible for the Section of Mechanical Engineering of such a Congress.

It was the sense of the Council that the same effort would be expended to better advantage if it were directed to make the regular meeting of the Society in June, partake of the character of an international reunion.

The Council has had before it during the year the provision of Article C52 concerning the organization of sections of the Society for purposes in harmony with its constitution and by-laws. Correspondence with the Secretary has been held concerning the organization of such sections in the cities of Cincinnati and Milwaukee. Other local organizations in various cities have also been formed during the year in which members of the Society are actively interested. The Council has referred the whole question of the organization of sections to a special committee consisting of Messrs. Swasey, Hutton, Hunt, Jesse M. Smith, and John W. Lieb, Jr.

This committee has not presented its report of recommendations.

Through the interested courtesy of Professor J. H. Barr, the Smith Premier Typewriter Company has presented to the Society a typewriter of its manufacture, together with a convenient and improved desk. Mr. H. H. Suplee and Mrs. R. H. Davis, with the co-operation of Professor John E. Sweet, have presented to the Society a photograph of the first "Straight Line Engine" which was exhibited at the Centennial Exposition in 1876. Mr. Charles V. Mapes, in co-operation with Mr. James M. Dodge, has presented to the Society a bust in plaster of the late John Ericsson, past member of the Society, with an appropriate pedestal.

Some interested members learning that an unusually excellent copy could be obtained of the life-sized portrait of James Watt, which had been copied from the original hanging in the National Portrait Gallery of London, have made this copy a gift to the Society, and it now hangs upon the walls of the auditorium.

In every case the Council has extended sincere votes of thanks to the respective donors.

Messrs. John E. Sweet and Thomas A. Edison have been elected Honorary Members of the Society, under the provisions of Article C17 of the Constitution.

Under the appointment of a joint library committee, with the other engineering societies, the following communication was received:

To the Councils of the
American Institute of Mining Engineers,
American Society of Mechanical Engineers,
American Institute of Electrical Engineers

Gentlemen: At a meeting of the Joint Library Committee held on Tuesday afternoon, May 17th, 1904, at the office of Mr. John Hays Hammond, No. 71. Broadway, New York City, it was resolved that the present status of affairs be reported to the Council of the respective societies.

After extended conference and consideration, two plans have presented themselves in connection with the proposed library:

First: The more comprehensive plan for a great Engineering and Scientific Library. To extend and supplement the present joint collection to realize this plan would require the expenditure of about $100,000; and would further, involve a continual outlay of $30,000 per annum for maintenance.

Second: The lesser plan of combining and continuing the existing collections of the several societies, numbering some thirty thousand volumes, conservatively valued at about $80,000, which would not contemplate any immediate large outlay for books; the maintenance of which would cost about $12,000 per year, or $4,000 for each of the affiliated societies.

While the Committee favors the first plan, it does not see its way clear at present to obtaining the necessary funds although it has not as yet abandoned hope of being able to do so; but in order to be enabled to make further progress, the Joint Library Committee has decided to recommend that it be authorized to proceed at present upon plan Second, and be empowered to take action involving the expenditure of $4,000 annually for each society for the maintenance of the joint library, and further, that, as an essential resolution to the execution of this plan, that the respective societies,

Resolved, That the books contributed by the several societies shall be combined to form a single library, but with provision whereby each volume will be so marked and registered as to indicate and preserve the title and ownership in the Society by which it is contributed.

For the Joint Library Committee,
(Signed) H. H. Suplee, *Secretary.*

After discussion the Council took the following action:

Resolved, That the Council of the American Society of Mechanical Engineers hereby approves the proposed combination in the Union Engineering Building of its Library collection with the similar collections of the other two Societies;

Provided, That the title and ownership of its collection, and of future additions thereto, by purchase from funds provided by this Society, or by donations through this Society, shall remain in it, and further;

Provided, That the provisions of any deeds of gift shall be observed, so far as they relate to the organization and conduct of the consolidated Libraries, and the title to the volumes therein included;

Resolved, That upon the combination of the several Libraries in the manner above set forth, this Society will provide annually one-third of the estimated sum necessary for the administration and maintenance of the consolidated libraries, this sum not to exceed a total of $12,000 per annum for all three Societies, or $4,000 for each Society, except by unanimous consent;

Provided, That any portion of the annual amount so contributed not required for the purposes of administration and operation shall be expended for the purchase of books and other library material under such rules as may be jointly adopted, and that the proportionate share of any amount so applied to the purchase of books derived from the contribution of this Society shall be expended only for such books, or other publications, as may be selected or approved by the Library Committee of this Society;

Resolved, That the books contributed by this Society shall be combined with the books of the other constituent Societies to form a single Library, which shall be administered as such.

The Council has approved the recommendation of the Meetings Committee and the spring meeting of 1905 will be held in the City of Scranton, Pa.

Under the provision of Art. 31 of the Constitution the Council would present the following list of those elected to membership during the fiscal year 1903-1904:

MEMBERS.

Allen, Benjamin T., Harrison, Pa.
Allen, Walter Morrison, Cleveland, O.
Aiken, Chas. Wilson, New York, N. Y.
Albert, Otto, Muskegon, Mich.
Albright, H., Fleetwood, N. Y.
Andrew, James D., New York, N. Y.
Angus, Joseph Johnson, Chicago, Ill.
Arnold, George, Brooklyn, N. Y.
Baker, John C. W., Naugatuck, Conn.
Bausch, Frederick Emil, St. Louis, Mo.
Bennett, Charles Wilbur, Elwood, Ind.
Bidle, William Sanford, Cleveland, Ohio.
Black, Edward S., Chicago, Ill.
Bloomberg, Jonas H., Mexico, Mexico.
Brown, Hugh T., Lexington, Ky.
Carse, John B., New York, N. Y.
Child, E. T., Jamaica Plain, Mass.
Clemens, Abraham Bowman, Scranton, Pa.
Colwell, James Van V., West New Brighton, S. I., N. Y.
Conrad, Ernest B., Bayonne, N. J.
Cooke, Fred W., Paterson, N. J.
Corbett, Robert Henry, Houghton, Mich.
Coster, Maurice, Paris, France.
Darlington, Philip Jackson, Bloomfield, N. J.
Davol, George Keen, San Francisco, Cal.
Diemer, Hugo, Lawrence, Kansas.
Donnelly, Wm. T., New York, N. Y.
Dunbar, William Otis, Altoona, Pa.
Duncan, J. D. E., New York, N. Y.
Edgar, Ellis F., New York, N. Y.
Ellicott, Edw. Beach, Chicago, Ill.
Elmes, Charles Warren, Chicago, Ill.
Fleming, Henry S., New York, N. Y.
Folger-Osborne, G. F., Bombay, India.
Foucard, Marcel L., Streatham Hill, Surrey, Eng.
Gardner, Thos. M., Ithaca, N. Y.
Gerhard, William Paul, New York City.
Gilbreth, Frank B., Boston, Mass.
Goddard, Arthur L., Orange, N. J.
Goddard, Dwight, Cleveland, Ohio.
Gray, Neil, Jr., Oswego, N. Y.
Greene, Frederick S., New York, N. Y.
Grossman, Albert, Brooklyn, N. Y.
Harper, Lewis E., Hyde Park, Mass.
Harrington, John L., West New Brighton, N. Y.

Harrington, Norman T., Detroit, Mich.
Harris, William Allison, Pittsburgh, Pa.
Haven, Henry Merritt, Somerville, Mass.
Hayward, Stirling F., New York, N. Y.
Helander, A. H., Pueblo, Colo.
Hess, Howard D., Lawrence, Kan.
Holdsworth, Fred Dean, Claremont, N. H.
Hood, Ozni Porter, Houghton, Mich.
Howe, Albert W., Morristown, N. J.
Howells, Charles, New York, N. Y.
Hulett, George Henry, Cleveland, Ohio.
Ingersoll, George T., Schenectady, N. Y.
Jenness, Charles Harvey, Chicopee, Mass.
Johnson, Werner, Lynn, Mass.
Jones, Benjamin Needham, Harrison, N. J.
Kelman, John H., Pittsfield, Mass.
Kenyon, Alfred Lewis, Lima, Peru, S. A.
Knox, Luther L., Pittsburgh, Pa.
Lafore, John Armand, Philadelphia, Pa.
Larkin, William Harrison, Jr., Boston, Mass.
Lenfest, Bertram A., State College, Pa.
Lincoln, Robt. B., Boston, Mass.
Lindstrom, Nils Oscar, Brooklyn, N. Y.
Loomis, Burdett, Jr., New York, N. Y.
Lord, John Edward, New York, N. Y.
Lucas, Henry Martin, Glenville, Ohio.
Macdonald, Duncan H., Milwaukee, Wis.
MacMurray, James T., Plainfield, N. J.
Massa, Robert Falconer, Chicago, Ill.
Meier, Konrad, New York, N. Y.
Merrywether, Geo. Ed., Bridgeport, Conn.
Metcalf, Leonard, Boston, Mass.
Mix, Edgar W., Paris, France.
Monteagle, Robt. Charles, Morris Heights, N. Y.
Moore, Albert B., Bridgeport, Conn.
Moore, Wm. E., Connellsville, Pa.
Muckle, John Seiser, Philadelphia, Pa.
Nicholson, Samuel T., Wilkes Barre, Pa.
Pattison, Frank A., New York, N. Y.
Peirce, Arthur W. K., Germiston, Transvaal, South Africa.
Powell, John Edgar, Washington, D. C.
Pritchard, W. S., Bronson, Mich.
Quirk, Wm. M., Richmond, Surrey, England.
Randall, Dwight T., Urbana, Ill.
Reid, Joseph, Oil City, Pa.
Reid, Marcellus, Cleveland, Ohio.
Rickey, Walter J., South Bend, Ind.
Rivett, Edward, Boston, Mass.
Robinson, Frank H., New York, N. Y.
Safford, Arthur Truman, Lowell, Mass.
Sage, Harmon M., Chicago, Ill.
Sanborn, Frank Edwin, Columbus, O.
Schulte, George Henry, Milwaukee, Wis.
Shepard, George H., Syracuse, N. Y.
Shepard, Louis A., Newark, N. J.
Slade, Arthur Jarvis, New York, N. Y.
Smith, Alton Lincoln, Worcester, Mass.
Smith, Edward Joseph, Passaic, N. J.
Smith, Persifor Frazer, Jr., Dennison, Ohio.
Sprado, Carl G., Milwaukee, Wis.
Stebbins, Theodore, Columbus, Ohio.
Street, Edgar L., New York, N. Y.
Stillman, George Frederick, Syracuse, N. Y.
Stivers, William Durell, Yonkers, N. Y.
Tandy, Harry, Kingston, Canada.
Taylor, Clarence Landfear, Alliance, Ohio.
Teele, Fred Warren, Trinidad, British West Indies.
Toltz, Max E. R., Montreal, Canada.
Trampe, J. Adams C. L. de, Philadelphia, Pa.
Walker, Gilbert Stoddard, Wheeling, W. Va.
Warg, Robert, Milwaukee, Wis.
Waterman, Chas., Boston, Mass.
Wells, Edward C., Quincy, Ill.
Wells, James Hollis, New York, N. Y.
Wolf, Frederick William, Chicago, Ill.

ASSOCIATES.

Aldcorn, Thos.,
New York, N. Y.
Beebe, Murray Charles,
Pittsburgh, Pa.
Bell, John E.,
Barberton, Ohio.
Caldwell, John R.,
Brooklyn, N. Y.
Dornin, George A.,
Saylesville, R. I.
Garred, Ulysses Anderson,
Anaconda, Montana.
Gillette, James Walter,
Phoenixville, Pa.
Hill, Harold Herbert,
Cleveland, Ohio.
Holz, Fred H., Jr.,
Cincinnati, Ohio.
Johnson, Warren,
New Orleans, La.
Judd, Horace,
Columbus, Ohio.
Lauer, Conrad N.,
Philadelphia, Pa.
Logan, James,
Worcester, Mass.
Loscher, Albert P.,
Toledo, Ohio.
Ludeman, Edwin Henry,
New York, N. Y.
Merrill, Josiah Leverett,
Pittsburgh, Pa.
Nelson, James William,
Brooklyn, N. Y.
Nichols, Wm. Westerfelt,
Philadelphia, Pa.
Norton, Frederick Lee,
Racine, Wis.
Noyes, Henry Taylor, Jr.,
Rochester, N. Y.
Obert, Casin Watson,
Brooklyn, N. Y.
O'Neil, J. G.,
Chicago, Ill.
Parkhurst, Frederick Augustus,
Portland, Me.
Peck, Chas. B.,
New York, N. Y.
Pell, David W.,
East Pittsburgh, Pa.
Pond, Henry Otis,
Tenafly, N. J.
Ransom, Allan,
Chicago, Ill.
Rippey, Samuel Howard,
Philadelphia, Pa.
Saldana, Eduardo E.,
San Juan, Porto Rico.
Saunders, Ernest W.,
Brooklyn, N. Y.
Stevens, Harold Luther,
Syracuse, N. Y.
Umstead, Chas. Henry,
Lawrence, Mass.
Van Winkle, Edward,
New York, N. Y.
Waddell, Chas. Edward,
Biltmore, N. Car.
Williams, George Whitney,
Chicago, Ill.
Woods, Frank F.,
Boston, Mass.

JUNIORS.

Anderson, Harold Bentley,
Cleveland, Ohio.
Atwood, George D.,
New York, N. Y.
Bailey, Ervin G.,
Fairmont, W. Va.
Barnard, Elmer E.,
Lynchburg, Va.
Barrow, Elbert S.,
New York, N. Y.
Bennett, George G.,
Claremont, N. H.
Berliner, Richard W.,
New York, N. Y.
Boughton, Judson H.,
New York, N. Y.
Brown, Edwin Hacker,
Minneapolis, Minn.
Buckingham, Henry Hine,
Ithaca, N. Y.
Canby, Harry Beaver,
Dayton, Ohio.
Case, Albert H.,
New York, N. Y.
Chasteney, Charles Dunton,
New York, N. Y.
Chatard, Wm. Miles,
Chicago, Ill.
Cobleigh, Henry Rice,
New York, N. Y.
Colwell, Chas. A.,
New York, N. Y.
Coombs, Howard A.,
Washington, D. C.
Diederichs, Herman,
Ithaca, N. Y.
Dreyfus, Theo. Frank,
New Castle, Pa.
Dudley, Samuel William,
New Haven, Conn.
Ehrmann, John P.,
Watervliet, N. Y.
Gamon, Ernest,
Long Island City, N. Y.
Gardner, Henry,
Concord, N. H.
Gillett, William Lowry,
New York, N. Y.
Glasgow, Carr Lane,
New York, N. Y.
Harvey, Richard P.,
Barberton, Ohio.
Hayley, Wm. P.,
Chicago, Ill.
Heaton, Herman Creel,
Cincinnati, O.
Helmes, M. J.,
Buffalo, N. Y.
Henes, Louis G.,
New York, N. Y.
Hodge, George Orvil,
Wilkinsburg, Pa.
Horton, William Henry,
Chicago, Ill.
Howlett, Lewis G.,
Bay City, Mich.
Jones, Jarrard E.,
Brooklyn, N. Y.
Jackson, Percy,
Macon, Ga.

Jordan, Wm. Allen, West New Brighton, N. Y.
Jump, Edmund P., Sparrows Point, Md.
Kasson, R. S., Sandy Hill, N. Y.
Katzenstein, Martin L., New York, N. Y.
Kelly, Gregory Cook, Germantown, Pa.
Kenney, Lewis Hobart, Philadelphia, Pa.
King, Roy Stevenson, Minneapolis, Minn.
Klain, Arthur W., Grove City, Pa.
Kleinhans, Frank B., Pittsburgh, Pa.
Knoop, Theodore Matthias, New Orleans, La.
Knowlton, Frederic Kirk, Auburn, N. Y.
Lockett, Kenneth, Chicago, Ill.
Lowe, Henry Leland, New York, N. Y.
Massie, James Hughes, Buffalo, N. Y.
Matthews, Fred Elwood, Boston, Mass.
Midgley, Frederick William, Yonkers, N. Y.
Moran, Harry Powell, Chicago, Ill.
Morgan, Llewellyn, Brooklyn, N. Y.
Morison, Geo. Abbot, Milwaukee, Wis.
Morrison, Hunter, Cleveland, O.
Murphy, John Killam, New Haven, Conn.
Nate, Emile Henry, Newark, N. J.
Newbury, George K., Cleveland, O.
Norris, Henry Lee, New York, N. Y.
Pettit, Frank, Sharpsville, Pa.
Pitkin, Joseph Lovell, New Orleans, La.
Powell, Edward Burnley, New York, N. Y.
Rantenstrauch, Walter, Ithaca, N. Y.
Reis, Leslie R., New York, N. Y.
Richards, Chas. Dexter, Detroit, Mich.
Richards, W. Allyn, St. Louis, Mo.
Rossberg, Chas. Andrew, New Haven, Conn.
Schaeffler, Simon, New York, N. Y.
Schlemmer, Edmund, Cincinnati, O.
Schuetz, Frederick F., New York, N. Y.
Scott, Earl Francis, Jackson, Tenn.
Sibley, Mark Miller, Detroit, Mich.
Sibson, Horace E., Philadelphia, Pa.
Slawson, Harry Harding, Passaic, N. J.
Snodgrass, John McBeath, Urbana, Ill.
Springer, John J., Cincinnati, Ohio.
Sponsler, Charles Frederick, Washington, D. C.
Stacy, Harry Winthrop, Springfield, Mass.
Staples, H. A., Brooklyn, N. Y.
Stevens, Robt. H., Brooklyn, N. Y.
Stone, Thos. W., Ft. Wayne, Ind.
Thatcher, Richard P., Pittsburgh, Pa.
Townsend, Harry P., New Britain, Conn.
Wachalofsky, Chas. John, Cleveland, Ohio.
Westerfield, George S., Houston, Texas.
Wettengel, C. Albert, St. Louis, Mo.
Whittemore, Herbert L., Claremont, N. J.
Wilson, Henry D., Pittsburgh, Pa.
Woldenberg, I., Tarrytown, N. Y.
Woodward, Robert Simpson, Cincinnati, Ohio.
Young, Charles Henry, Fort Wayne, Ind.

The losses by death from the list of membership during the current year, have been as follows:

J. M. Allen, J. N. Barr, R. C. Blackall, R. M. Blankenship, H. M. Boies, Jas. T. Boyd, W. P. Canning, Jas. A. Connell, R. W. Davenport, S. C. Dinsmore, Geo. E. Dixon, Carl F. Eicks, Chas. F. Elmes, Clark Fisher, D. R. Fraser, Frank Kempsmith, F. H. Laforge, David J. Matlack, Jno. R. Matlack, Jr., F. de V. Olmsted, Alex Pollock, Clyde Randolph, Geo. Richmond, J. A. Roche, J. R. Slack, Henry I. Snell, W. G. Vernon, Wm. Wallace, Herman E. Witte.

The Council would call particular attention to the reports of its Standing Committees which appear as appendices to this report:

APPENDIX I.

REPORT OF THE FINANCE COMMITTEE.

The Finance Committee presents the following report to the Council.

Under the duties prescribed by the Constitution and By-laws the Finance Committee at an early meeting arranged the details for the financial system of accounts to be used during the year. A Sub-committee was appointed under whose direction forms of vouchers were prepared and all the expenditures of the year have been carried on under these forms. The system has provided that each of the Standing Committees makes a requisition for the sum to be expended for the purposes of each Committee and specifying the object for such expenditure. The Secretary of the Society in carrying out the orders of the Committees returns the voucher to the Chairman of the Committee and on his approval, in accordance with By-law B21, it passes to the Finance Committee who audits the bill for payment.

The Committee submits the financial statements hereto appended, which have been prepared by the accountant of the Society, and submitted to the Audit Company of New York for scrutiny and audit. Their report is appended. The Financial Statements include:

Sheet A, a Statement of Cash Account, Receipts and Disbursements.

Sheet B, a Statement of Income and Expenditure.

Sheet C, a Balance Sheet of Assets and Liabilities.

Sheet D, a Tabular Statement of Changes in these Assets and Liabilities as compared with the condition September 30, 1903.

The Committee submits also a Sheet of estimated receipts and expenditures for the coming year, 1904–5, based on the assumption that no important changes are made in the direction or extent of expenditure for the coming year as compared with the year which has closed. This estimate is designated Sheet E.

In comment thereon the Committee would ask that the Council consider from the report of the accountant and the audit of the Audit Company of New York, the consequence of the resolution of the Council whereby ninety per cent. of the receipts and initiation fees each year is to be set aside to the credit of a reserve fund. It will be recalled that the entire amount of the special subscriptions received in cash from subscribers to the fund of the Library Association are to be set aside as not available for the expenses of the Society. One per cent. of the cash receipts from dues are to be set aside to develop the library.

It has been possible since the passage of this resolution to credit a cash asset each year to offset the liability for the trust fund of the library and set aside the entire amount of the life membership fees. These reserves are in the savings bank. The income of the past two years, however, has not been such as to enable the Society to credit a cash asset sufficient to cover the liability of ninety per

cent. of the initiation fees. Last year the liability which was not coverable from cash receipts, September 30, 1903, was $3,974. Ninety per cent. of the receipts from initiation fees this year, is $5,292; it has been possible to set aside $1,900 from initiation fees, which has been used to reduce the liability of the last fiscal year. This makes the book liability to this account, foot up to $2,917.50. The liability to the Library Development Fund of one per cent. of the receipts from dues is this year, $344.87, for which there has been no cash income to offset. It will be apparent therefore, that unless the income grows rapidly in relation to the expenditure, the Society will not be able to deposit in savings banks sufficient funds to cover fully these reserved funds.

The Committee would submit also computations which have been deduced from the accounts of the current fiscal year, as follows:

(1) Total members as per July, 1904, catalogue	2,740		
Add new members who have paid since then	7—	2,747 00	
Deduct for members who have paid no dues; Life Members	108		
Deaths and resignations, without payment	6		
Lapsed memberships	25		
Members who have not paid current year at September 30, 1904	211—	350 00	
Paying membership, 1903–1904		2,397 00	
(2) Total income exclusive of 1 per cent. from dues carried to Library Development Fund, 90 per cent. from initiation fees, entire life membership receipts, carried to Reserve Fund, and entire Sinking and Fellowship Fund, subscriptions of Mechanical Engineers' Library Association		$41,520 73	
Income per paying member (computed)		17 44	
Income per paying member, from dues only (computed)		14 91	
(3) Total expense incurred year October 1, 1903, to September 30, 1904, less cost operating house ($3,720.12), mortgage interest ($1,402.50), repairs and renewals ($1,561.30), depreciations house and furniture ($444.43)—$36,-397.55:			
(4) Total expense incurred for publications, October 1, 1903, to September 30, 1904		14,919 22	
(5) Total expense incurred for salaries in Society's office same period		9,570 00	
(6) Total expense incurred for all other accounts except house		11,908 33	$36,397 55

Amount brought forward		$36,397 55
(7) Total expense incurred for house, including interest on mortgage, repairs and renewals and depreciations	7,128 35	
Deduct income earned from rent of rooms and hall	1,354 50—	5,773 85
Net expense incurred for year 1903–1904...		$42,171 40
(Gross expense, $43,525.90 less rental income $1,354.50, equals $42,171.40).		

Expenses incurred per paying member, October 1, 1903, to September 30, 1904:

(8) For all purposes including house	$17 59	
(9) For house operation including interest and repairs and depreciations	2 41	
(10) For all purposes exclusive of house	$15 18	
(11) For publications, printers' work, engraving, binding and distribution $6 25		
(12) For salaries in Society's office..... 3 95		
(13) For all other expenses except house ..... 4 98—	15 18	
(14) For house operation exclusive of mortgage interest, repairs and renewals and depreciations ..	98	
(15) For house operation exclusive of mortgage interest, but including repairs, renewals and depreciations	1 92	
(16) For operating library	38	
(17) For Postage, circulars, catalogues, and stationery and printing in Society's office.....	3 01	
(18) For meetings, and all other expenses not otherwise allotted above.....	1 55	

Comparative income earned with expense incurred per paying member:

* Income earned from all sources per paying member, per (2)	$17 44
Income earned from dues only per paying member, per (2)	14 91
(19) Excess expense incurred all purposes, per paying member over income earned from all sources, per paying member	15
(20) Excess expense incurred all purposes, per paying member over income earned from dues alone, per paying member	2 68

From the above it would appear that the Income from dues per member, $14.91, is less than the expenditure per member, $17.59. This indicates clearly

* Only 99 per cent. of dues and 10 per cent. of Initiation Fee receipts considered as income. Receipts for Life Memberships and subscriptions to Fellowship and Sinking Funds not considered as income.

the demand which the Standing Committees have had to make upon the receipts from initiation fees, mainly to meet the expenditure asked for by the Standing Committees of the Society and approved by the Council. The Committee believes that the policy of spending receipts from initiation fees, to the proportion required for the present year, is not a good financial procedure, since initiation fees may fall off through many causes outside of the Society, which would diminish the number of applications for membership. The phenomenal growth of the Society and the unusual sums received from initiation fees should be noted in connection with the sheets of accounts presented.

The Finance Committee has also been made by Resolution of the Council to be the Committee which shall audit expenditures in those directions of the Society's work which were not specifically assigned to another Standing Committee under the provisions of the Constitution. The expenditure under these heads is ordered specifically by the Council itself by Resolution, but the Finance Committee is to audit the bills therefor.

These expenditures include the salaries of the Secretary and of the office staff, and the expenditure for miscellaneous accounts passing through the administrative office of the Society. The elements of these two channels of expense have been as follows:

Salaries:

Secretary	$3,600 00
Assistant to Treasurer and Accountant	2,400 00
Assistant to Secretary	1,600 00
Stenographer	840 00
Stenographer	410 00
Mail Clerk	720 00
	$9,570 00

Certificates and introduction cards	$238 61
Badges, distribution and repairs	36 62
Circulars	2,311 57
Meetings	2,235 36
Year Books and Pocket Lists	3,198 75
Office accounts, exclusive of salaries	1,756 75
Headquarters, St. Louis Exposition	341 85
Expert fees, examination of accounts	150 00
Legal Expenses	50 00

In comment thereon, the Committee would note that the circulars connected with the process of admission into the Society, and which are designated as "admission circulars," have cost $836.84; miscellaneous printing in the form of circulars in connection with the meetings of the Society, programs and the like, have cost $867.68; circulars which are required for general purposes, miscellaneous announcements and the like, $607.05. The expenses for the two general meetings in New York and in Chicago amounted to $2,077.68; the expenses for the Tuesday evening meetings in New York City, which were paid from the Society's funds, amounted to $157.68.

By direction of the Council the list of members of the Society, was changed slightly in form and received the designation of its "Year Book." The repro-

ductions of the portraits of members of the Council were incorporated into this publication and a special design of cover was ordered this year for the first time. A list of members, bound in convenient "vest pocket size," is issued in July, which includes additions and corrections which have accumulated during the six months subsequent to the issue of the Year Book in January.

The Committee takes this occasion also to call the attention of the Council to the fact that by the provisions of the agreement which the Society expects to enter into with the other Societies participating in the gift of a building by Mr. Andrew Carnegie, the Finance Committee will be called on on or before the first day of July, 1905, to provide for a payment to the fund of that Committee of the sum of $8,000 in cash. The Committee have made this payment an item in the estimate of expenditures for the year and call the attention of the Council to the obligation imposed by this action.

The Council also directed that the Society maintain in the city of St. Louis, on the Exposition grounds of the Louisiana Purchase Exposition, a headquarters for the convenience of members of the Society who should visit the Machinery Hall of that exposition. Very minor expenses were incurred for furnishing and putting that headquarters into convenient condition for use, but the Committee has had to meet so much of the salary of the Society's representative in that headquarters as has accrued during the period of the Exposition, up to September 30, 1904. The balance will have to be met in the next fiscal year.

Respectfully submitted.

CHAS. WALLACE HUNT,
DAVID TOWNSEND,
MILTON P. HIGGINS,
FRED W. TAYLOR,
STEVENSON TAYLOR,
Finance Committee.

APPENDIX II.

REPORT OF COMMITTEE ON MEETINGS.

The Committee on Meetings would report that pursuant to the provisions of the Constitution and By-laws, the Committee organized by electing Mr. H. G. Reist Chairman.

The Committee has had before it the preparation of the program of the joint meeting of the American Society with the Institution of Mechanical Engineers, held in Chicago, and the preparation of the program for the annual meeting.

It was the belief of the Committee that for such joint meeting the policy should be followed of inviting papers on selected topics rather than the plan of following the lead of contributing members. To this end it aimed to secure papers on the "Steam Turbine," on the "Gas Engine," and the "Machine Tool for Rapid Production on a Large Scale," and on the performance of the "Modern American Locomotive." Some of these topics it was found impossible to secure, and the Committee completed its program by selecting from other directions.

For the Chicago Meeting, seven papers were contributed by the Institution of Mechanical Engineers, and both the American and British papers were discussed by members of both societies. The program for entertainment, provided by the courtesy of the Chicago members, was most enjoyable, as will be seen by reference to the minutes of the meeting reported in the Proceedings.

The Committee on Meetings has also commended to the Council that the spring meeting of 1905 be held in Scranton, Penna., pursuant to an invitation received from that city.

Respectfully submitted,

H. G. REIST,
ARTHUR L. WILLISTON,
W S. ACKERMAN,
H. DE B. PARSONS,
WALTER MCFARLAND,
Committee on Meetings.

APPENDIX III.

REPORT OF COMMITTEE OF PUBLICATION.

The Publication Committee presents the following report of matters under its direction.

At the close of the last fiscal year it was estimated that the sum of $6,300 would be required to complete Volume XXIV. The amount actually required has proved to be slightly greater, viz.: $6,522.50, making, with the expenditure in the previous year, the total cost of the volume $15,179.24. It contains 1,563 pages, and the total cost per copy was $5.62.

The *Transactions* for the current year, Volume XXV., will contain the proceedings of the New York and Chicago meetings. Its issue has unavoidably been delayed to permit revision by participants in the Chicago meeting who are members of the Institute of Mechanical Engineers of Great Britain. The amount expended for this volume to the end of the current fiscal year amounts to $8,309.22. It is estimated that in round numbers $6,700 will be needed to complete the volume, in which event its total cost will be about $15,000.

The items composing the above expenditure of $8,309.22, which technically has been incurred under the direction of the Publication Committee, and the total of each item are as follows:

Advance papers	$2,863 82	
Revised papers	564 30	
Stenographer's fees	289 00	
Engraving	1,481 17	
Composition and electrotyping	2,573 66	
Binding extra copies	179 10	
Postage and express	73 92	
Storage, including insurance	277 50	
Boxing plates	6 75	
Total		$8,309 22

The estimated amount required to complete Volume XXV., is in detail as follows:

Revised papers, Chicago meeting	$900 00	
Composition and electrotyping	2,100 00	
Binding	2,500 00	
Distribution expenses, express	1,100 00	
Boxing plates	10 00	
Total reserved to complete Volume XXV.		$6,610 00
Estimated Total Cost, Volume XXV.		$14,919 22

The Publication Committee refers to the Council the following matter for its consideration.

In order to reduce the expenditure for publications during the year 1902–3 as much as possible, the issue of Volume XXIV., printed originally from the plates was reduced to 2,700 copies. The growth of the Society during the last eighteen months has been unprecedentedly large, and a considerable proportion of the new members are either entitled to receive Volume XXIV., or likely to wish to purchase it. At this date the edition is practically exhausted. The printers estimate that an edition of 250 copies, 50 of them in standard binding and the remainder unbound, will cost $740. The Secretary recommends the printing of this supplemental edition, and the committee refers the question to the Council. If approved, the above amount, $740, should be included in the budget for the next fiscal year.

The Committee recommends a change in the present practice in regard to action by the Committee on matter proposed to be included in the *Transactions*. Unless the Publication Committee is selected from among the members in New York and vicinity (which is not desirable) it is practically impossible to get them together at the dates and for the time needed for the performance of their duties. Recognizing this fact the Secretary has sent duplicate copies of all proposed matter to each member of the Committee, at the same time stating that, unless objection was made within a specified time, it would be assumed that each of the members approved the matter as presented. The time allowed for the examination of the matter so referred to the members of the Committee has been too short to permit of careful review of the numerous papers. Therefore the Committee recommends that while the present method be continued a more liberal allowance of time should be made in which the members of the Committee may examine the papers, communicate with one another if desired, and report their conclusions. Sixty days would not be unreasonable time for this purpose, but, if this is too long, the allowance should be liberal as other conditions permit.

Respectfully submitted,

HENRY R. TOWNE, Chairman,
HENRY SOUTHER,
GEORGE M. BASFORD,
C. J. H. WOODBURY,
HENRY H. SUPLEE,

Committee on Publication.

APPENDIX IV.

MEMBERSHIP COMMITTEE.

The Membership Committee would report that it has held frequent meetings during the year for the consideration of applications for membership, and has received and considered 392 such applications, representing the largest accession to the membership list in any one year in the history of the Society.

At its first session the Committee recommended (and the Council subsequently approved) a system of modified blank forms which have since been in use.

It has been the effort of the Committee so to interpret the wording of the Constitution that while its intent was steadily kept in mind, the standard of professional achievement exacted from applicants should be firmly maintained at a high level, so that the value of membership in the several grades might be proportionately enhanced. As the result of this policy many applicants have been graded to a grade lower than that applied for

The Committee ask that the members who propose new candidates, and who subsequently fill out the blank forms concerning them, should be definite and positive in opinion and statement, so that the Committee may be rightly guided in the discharge of its duties.

The combined ballot and professional service sheet, which was adopted on the recommendation of the Committee seems to have been well received.

Respectfully submitted,

R. H. Soule, Chairman,
Francis H. Stillman,
Ira H. Woolson,
Wilfred Lewis,
Jesse M. Smith.

APPENDIX V.

REPORT OF LIBRARY COMMITTEE.

The Library Committee presents the following report. The Library has been open every day between the hours of 10 A.M. and 10 P.M., except on Sundays, Thanksgiving, Christmas and New Year, and excepting the month of August, when it was closed for cleaning and overhauling.

There have been 3,050 visitors to the Library during the twelve months, or an average of ten persons per day. The use of the Library during the evenings has been particularly satisfactory. The number of volumes in the Library September 30, 1904, is as follows:

Books	8,782
Pamphlets	3,952
Maps	33

The book value of the Library at the same date is $11,884.52, which is the sum of the valuation at the end of the previous year's report, and of additions made during the year.

The additions to the Library in the form of exchanges which have been received as the equivalent of the annual volume of the Society's Transactions, have amounted to $740.40.

The Committee has expended for the purchase of books during the year, $164.60, making the total amount expended for additions to the Library for the year, $905. It should be observed, however, that this is not all in the form of direct cash expenditure. There has been an expenditure of $282.40 for binding periodicals and pamphlets received in exchange from other organizations or publication agencies. There remains standing on the Society's books a charge against the publishing house of Van Nostrand & Co., for *Transactions* furnished them for which the Society is to purchase books from them, amounting to $241.75. With the house of Spon and Chamberlain there is similar balance in our favor of $16.50.

For the conduct of the work in the Library, Miss Thornton is employed, and in addition the service of other members of the staff when they can be spared from other duties.

The manuscript of the card catalogue has been extended, and so far as the book titles in the library are concerned, has been completed. The Committee hopes in the near future to issue in printed form a library catalogue for distribution to the membership. This has not been done up to the present time on account of the expense involved, and also because joint action is under advisement with similar committees of the Institution of Mining Engineers, and the Institution of Electrical Engineers, looking to a common policy of administration of their respective libraries during the years which intervene before the Carnegie Library of Engineering shall be available in the new house of the Societies.

Under the direction of this conference Committee, the list of exchanges, periodicals, journals and the like has been printed and will doubtless be ready for distribution in the near future. The Committee have hesitated to incur the expense of the catalogue of our library alone, if by the changes of the next few years it would be made either valueless or unnecessary.

Respectfully submitted,

STEPHEN W. BALDWIN, Chairman,
E. A. UEHLING,
W. D. FORBES,
FRED. J. MILLER,
FRED'K M. WHYTE,
Library Committee.

APPENDIX VI.

REPORT OF THE HOUSE COMMITTEE.

The Committee intrusted with directing the expenditure for the maintenance of the house of the Society, which is its office headquarters also, and which provides for certain rooms in the upper floors at the service of visiting members for short stays in the city, would report as follows. The expenditures under

the direction of the Committee on requisitions during the year have been as follows:

For operating expenses	$3,720 12
For interest on purchase mortgage	1,402 50
For repairs and renewals	1,561 30
	$6,683 92

The most serious element of expenditure under the Committee's hand for the year has been the item of painting and papering the upper part of the house—$591.30.

During the two previous years there has been little or no expenditure in this direction and the house has been deteriorating under the restricted expenditure for maintenance. The Committee had also to meet considerable repairs to the heating apparatus, put in two years before, amounting to $192.90.

The Committee report that the receipts which offset the expenditures on account of the house, from rental of its auditorium to other societies and from the use of its upstairs rooms, has been less this year than hitherto. This fact is to be explained by the growth of those organizations which have been using the hall, which has kept pace with the growth of the Society itself. The consequence of such growth is that the auditorium has become inadequate for their needs and they have therefore sought larger accommodations and withdrawn their financial support to the hall in this way.

The Society of Naval Architects and Marine Engineers continues to meet in the auditorium, but the Institute of Electrical Engineers, and the Society of Heating and Ventilating Engineers, have both felt compelled to move elsewhere. The following table shows the expenses incurred in detail for the house:

Interest on mortgage	$1,402 50
Gas and electric light	689 80
Fuel	365 00
Janitor's supplies	201 83
Laundry	393 58
Insurance	118 17
Repairs and Renewals, house	1,094 16
Repairs and renewals, furniture	467 14
Wages	1,766 65
Incidentals	185 09
Total exclusive of depreciation	$6,683 92
Depreciations	$444 43

The receipts from rentals, subtracted from the expenditure, makes the net expenses of operating the house, $2,365.62. If to the operating expenses be added the interest on mortgage, repairs and renewals, rentals and depreciations, but excluding an interest on the value of the equity, the total expense is $7,128.35.

The Committee would call attention to the notable increase in the cost of lighting for the current year, as compared with the year preceding. This is partly due to the completion on the north side of the street, in which the house is situated, of two high buildings, which materially darken the working offices.

This has compelled much greater use of artificial light than heretofore. The electric meter is also charged with errors in registry, which are now under advisement with a view to securing a rebate if the amounts prove to be in excess of current actually used.

The House Committee employs for the conduct of the house, a Janitor at $45, Assistant Janitor, $40, and a maid, $20.

Respectfully submitted,

CARLETON W. NASON, Chairman,
RALPH L. MORGAN.
THOS. R. ALMOND,
JOHN C. KAFER,
Committee on House.

APPENDIX VII.

AUDIT OF SOCIETY ACCOUNTS.

The Finance Committee would report that they caused an audit of the books of the Society for the fiscal year 1903-4 to be made by the Audit Company of New York, and the Report of said Audit Company follows:

THE AUDIT COMPANY
OF NEW YORK.
Cedar & William Streets.

CHARLES WALLACE HUNT, ESQ.,
Chairman Finance Committee,
American Society of Mechanical Engineers,
12 West 31st St., New York City.

Dear Sir:

Agreeably to your request, we have made an audit of the books of your Society for the year ended September 30, 1904, and enclose herewith the printed accounts duly signed.

We have pleasure in stating that in our opinion the form of the Balance Sheet as drawn up by the Society's Accountant, Mr. Francis W. Hoadley, and which will be presented to the members, is excellent, and that it reflects credit on that gentleman. No difficulty was experienced during the course of our examination, and everything that our representative asked for was duly placed before him, for which we desire to express our obligations.

In conclusion, we certify that the figures contained in the various documents submitted are correct, and that the signed Balance Sheet accurately shows the true financial condition of the Society as at September 30, 1904.

Very truly yours,

THE AUDIT COMPANY OF NEW YORK.

E. T. PERINE,
General Manager.

New York, October 27, 1904.

SHEET A.

COMBINED STATEMENT OF CASH ACCOUNT.

FISCAL YEAR, 1903–1904.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

September 30, 1903, *to October* 1, 1904.

Receipts.

1903. Oct. 1.	To Cash on hand		$85 73
1904. Sept. 30.	" Cash receipts other than Trust Fund Subscriptions during year		46,575 57
"	" Cash receipts, Trust Funds of Mechanical Engineers' Library Association, Fellowship and Sinking Funds of said Association		398 00
"	" Cash received by legacy from Geo. W. Weeks		1,957 00
			$49,016 30
1904. Oct. 1.	To Cash in East River National Bank, New York		$84 66

Disbursements.

1904. Sept. 30.	By Disbursements for Expenses of Fiscal Year 1902–3 (see Sheet C, Surplus)	$6,522 50	
"	" Disbursements for Expenses of Fiscal Year 1903–4, all bills received to date paid	37,554 14	
	Total Disbursements for Year		$44,076 64
"	By Money deposited in Savings Banks:		
	Trust Fund—M. E. L. A.	$398 00	
	Reserve Fund—A. S. M. E.	2,500 00	
	Geo. W. Weeks Legacy Fund	1,957 00	
	Totai Amount Deposited in Savings Banks.		4,855 00
			$48,931 64
"	By Cash in East River National Bank, New York		84 66
			$49,016 30

SHEET B.

COMBINED STATEMENT,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

INCOME AND EXPENSE ACCOUNT, *September* 30, 1904. FISCAL YEAR, 1903–1904.

Income Earned.

Dues Account—		
Dues collected—Fiscal Year 1903–4	$33,082 52	
Outstanding and considered good, Fiscal Year 1903–4	2,882 48	
	35,965 00	
Less—1 p. c. total cash collections carried to Reserve Fund—A. S. M. E.	344 87	
		$35,620 13
Life Membership	600 00	
Less—100 p. c. carried to Reserve Fund—A. S. M. E.	600 00	
		0 00
Initiation Fees	5,880 00	
Less—90 p. c. carried to Reserve Fund—A. S. M. E.	5,292 00	
		588 00
Sales Account (less allowances and goods returned)—		
Publications, except those furnished exchanges for books for the Library	3,196 54	
Publications — Transactions furnished exchanges for books and papers for the Library	740 40	
		3,936 94
Sales of Electros—Net Gain		5 66
Sales of Second Hand Vols.—Net Gain		15 50
House Account, Rent—		
For Sleeping Rooms	1,229 50	
" Hall	125 00	
		1,354 50
Total current income		$41,520 73
Balance—Excess Expenses incurred over Current Income for Fiscal Year 1903–4 charged against Surplus Account		2,005 17

Expenses Incurred.

Transactions, including Distribution—			
Total Disbursements—Fiscal Year 1903–4		$14,831 72	
Less—Amount expended to complete Vol. XXIV., the volume for 1902–3, charged against Surplus at Sept. 30, 1903	$6,300 00		
Add excess of actual amount expended over above amount provided for, this amount charged against Surplus in year 1903–4	222 50	6,522 50	
Paid on account of Vol. XXV—Fiscal Year year 1903–4		$8,309 22	
Add—Estimated amount required to complete Vol. XXV. for Fiscal Year 1903–4, including cost of distribution		6,610 00	$14,919 22
Office Account—			
Expenses other than Salaries—			
Stationery and Printing		$524 07	
Postage, general		623 16	
Telegraph, telephone and messenger		94 09	
Supplies		268 20	
Incidentals		247 23	
		$1,756 75	
Salaries		9,570 00	11,326 75
Meetings—			
Annual		$772 29	
Spring		1,305 39	
Monthly		157 68	
			2,235 36
Committee Work			14 10
Year Books and Pocket Lists of Members—			
Year Book—Jan., 1904, Composition, Presswork and Paper		$1,755 79	
Pocket List, July, 1904, Composition, Presswork and Paper		1,052 50	
Distribution on both—postage		390 46	
			3,198 75
Admission		$836 84	
Meetings		867 68	
General		607 05	

Presswork and Paper	1,052 50	
Distribution on both—postage	390 46	
		2,495 76
Admission	$836 84	
Meetings	867 68	
General	607 05	
		2,311 57
Library—		
Binding Books	$282 40	
Expense	56 40	
Salary, Librarian	600 00	
		938 80
Certificates and Membership Cards—		
Certificates	$218 01	
Cards	20 60	
		238 61
Distribution and Repair of Badges—		
Repairs	$9 00	
Distribution	27 62	
		36 62
Legal Expenses and Expert Fees—		
Legal Expenses		50 00
Expert Accountant, Auditing Books		150 00
Uncollectable Accounts—Written off		21 90
House Account—		
Lighting	$689 80	
Fuel	365 00	
Janitor's Supplies	201 83	
Laundry	393 58	
Insurance	118 17	
Wages	1,766 65	
Incidentals	185 09	
Total Cost of Operating House	$3,720 12	
Interest on Mortgage of $33,000 @ 4½ p. c.	1,402 50	
Repairs and Renewals—		
House	$1,094 16	
Furniture	467 14	
	1,561 30	
Depreciations—		
House Furniture	$120 43	
Heating and Vent. App.	324 00	
	444 43	
Total Expense of House exclusive of Interest on Value of Equity		7,128 35
Headquarters, St. Louis Exposition		341 85
Decreased Stock of Transactions on Hand—		
Inventory, Sept. 30, 1903	$15,607 00	
Less—Inventory, Sept. 30, 1904	14,992 98	
		614 02
		$43,525 90

$43,525 90

SHEET C.

COMBINED BALANCE SHEET.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

September 30, 1904.

Assets.

Property— 12 W. 31st Street, N. Y. Value Appraised	$85,000 00	
Less Mortgage	33,000 00	$52,000 00
Fixtures and Furniture—Heating and Ventilating apparatus	$3,240 00	
Less 10 p. c. Depreciation for 1903–4	324 00	
Book Value, Sept. 30, 1904	2,916 00	
House Furniture—Book Value, Oct. 1st, 1903	$1,204 26	
Less 10 p. c. Depreciation for 1903–4	120 43	
	$1,083 83	
Add for New Furniture Purchased 1903–4	277 89	
Book Value, Sept. 30, 1904	1,361 72	4,277 72
Library—Books, Pamphlets, etc.— Book Value, Oct. 1st, 1903	$10,979 52	
Additions during year	905 00	
Book Value, Sept. 30, 1904		$11,884 52
Stock of Transactions—Inv. Sept. 30, 1904		14,992 98
Stock of Badges—Inv. Sept. 30, 1904		153 00
Arrears of Dues—Fiscal Year 1903–4		2,882 48
Arrears of Trust Funds—Fiscal Year 1903–4		
Sinking	23 00	
Fellowship	10 00	33 00
Insurance Premiums Paid in Advance		87 46
Sundry Debtors—Due the Society:		
For Volumes and Pamphlets	415 00	
" Room Rent	36 50	
" Badges	4 50	
" Electros of Cuts	44 15	
" Office Expenses—Telephone Tolls	75	

Liabilities.

Reserved for Uncompleted Work on Volume XXV. for Year 1903–4 —		
Composition and Electrotyping	$2,100 00	
Binding—Paper Covers	100 00	
" Leather Covers	2,400 00	
Boxing Plates	10 00	
Distribution—Postage and Express	1,100 00	
Revised Papers	900 00	$6,610 00
Reserved for Interest on Mortgage Accrued		350 62
Advance Payments		162 45
†Library Development Fund—Principal	664 26	
* " " " Interest	5 58	669 84
**Reserve Fund*—Life Membership	1,000 00	
* " " Initiation Fees	2,900 00	
* " " Interest on Life Membership Fund	7 00	
**Reserve Fund*—Interest on Initiation Fee Fund	17 50	3,924 50
‡*Trust Fund, M. E. L. A.*—Principal	1,912 23	
* " " " Interest	25 92	1,938 15
Altoona Mech. Library—Credit Balance with the Society		18 00
*Geo. W. Week's Legacy Fund		1,957 00
Surplus, Sept. 30, 1903	74,679 40	
Deduct—Excess Actual Expenditure to Complete Vol. XXIV. for Year 1902–3.		
Reserved to Complete Vol. XXIV	$6,300 00	
Actual Expenditure for same	6,522 50	
Excess Actual Expenditure over Amount Reserved now charged against the Surplus at end of that year as a portion of the Expense of that year	222 50	

Sundry Debtors—Due the Society:			
For Volumes and Pamphlets	415 00		
" Room Rent	36 50		
" Badges	4 50		
" Electros of Cuts	44 15		
" Office Expenses—Telephone Tolls	75		
		500 90	
Suspense Account (over-due accounts)—			
For Room Rent	2 00		
" Publications and Pamphlets	15 60		
" Electros of Cuts	23 10		
		40 70	
D. Van Nostrand Co.—Exchange Account: Balance due the Society for Publications		241 75	
Spon and Chamberlain— Balance due the Society for Publications		16 50	
Stock Second-hand A. S. M. E., Transactions— On hand—Inventory, Sept. 30, 1904		126 00	
Union Dime Savings Institution, New York Trust Fund, M. E. L. A.—On deposit—			
Principal	$1,879 23		
Interest	25 92		
	1,905 15		
Institution for Savings of Merchants Clerks, New York Reserve Fund, A. S. M. E.			
Life Membership—Principal	$1,000 00		
" " Interest	7 00		
Library Devl. Fund—Principal	319 39		
" " " Interest	5 58		
Initiation Fees—Principal	2,900 00		
" " Interest	17 50		
	4,249 47		
Geo. W. Week's Legacy—Principal	1,957 00		
Total Funds in Savings Banks		8,111 62	
Cash—East River National Bank, N. Y.		84 66	
		$95,433 29	

charged against the Surplus at end of that year as a portion of the Expense of that year	222 50	
	$74,456 90	
Deduct—Excess of Expenses incurred over Current Income for Fiscal Year 1903–4 (see Sheet B.)	2,005 17	
	$72,451 73	
Add—Amount of credit to Reserve Fund, Initiation Fees not covered by Cash in Savings Bank, 1902–3, now written off	2,059 00	
Add—Amount of credit to Reserve Fund, Initiation Fees for 1903–4 which could not be covered by cash deposit in Savings Bank now written off	5,292 00	
Surplus, September 30, 1904		79,802 73
		$95,433 29

* Covered in full by cash in Savings Bank (see Assets).

† Against this there is $319.39 on deposit in Savings Bank (see Assets), same being the amount credited to this account at September 30, 1903 in full, the income for the fiscal year, 1903–4, not allowing sufficient funds to enable any cash amount to be created to cover the $344.87 credited to it for 1903–4.

‡ Against this there is $1,879.23 on deposit in Savings Bank (see Assets). The balance $33.00 being amount outstanding uncollected at September 30, 1904, but considered as collectable (see Assets " Arrears Trust Funds, 1903–4 ").

Having audited the above Balance Sheet and accompanying Accounts, prepared by Mr. Francis W. Hoadley, with the books and vouchers of the Society, we certify that the same, in our opinion, fully and fairly represents the condition of the Society at September 30, 1904.

THE AUDIT COMPANY OF NEW YORK,
E. T. PERINE, *Gen'l Manager.*

NEW YORK, *October* 27, 1904

TABULAR STATEMENT OF CHANGES IN ASSETS AND LIABILITIES,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION COMBINED.

September 30, 1904. *compared with September* 30, 1903,
and remarks in comment thereon.

Designed and compiled by FRANCIS W. HOADLEY, Assistant to Treasurer, and Accountant. Showing whence arise the gains and losses in the financial condition of the Society, as affecting its Assets and Liabilities.

Accounts.	Assets and Liabilities.				Losses and Gains.	
	September 30, 1903.		September 30, 1904.		During Year 1903-4.	
	Assets.	Total.	Assets.	Total.	Losses.	Gains.
Property, 12 W. 31st Street, Equity	$52,000 00		$52,000 00			
Fixtures and Furniture—						
Heating and Ventilating Apparatus, Book Value	3,240 00		2,916 00		*$324 00	
House Furniture, Book Value	1,204 26		1,361 72			†$157 46
Library—Books, Pamphlets, etc., Book Value	10,979 52		11,884 52			905 00
Stock of *Transactions* on hand, Inventory Value	15,607 00		14,992 98		614 02	
Stock of Badges on hand, " "	90 00		153 00			63 00
Arrears of Dues	1,395 25		2,882 48			‡1,487 23
" " Sinking Fund	10 00		‡23 00			‡13 00
" " Fellowship Fund	5 00		‡10 00			‡5 00
Insurance Paid in Advance	144 13		87 46		56 67	
Due from Sundry Debtors—Publications	121 80		§415 00			293 20
" " " Hall Rent	35 00		.....		35 00	
" " " Room Rent	69 50		36 50		33 00	
" " " Badges	9 00		4 50		4 50	
" " " Electros	23 10		44 15			21 05
" " " Miscellaneous	7 00		75		6 25	
Suspense Account (Overdue)—Room Rent	.....		2 00			2 00
" " " Publications	12 90		15 60			2 70
" " " Electros of Cuts	.....		23 10			23 10
D. Van Nostrand Co.,—Exchange Acct., Balance due Society	328 75		241 75		87 00	
Engineering Magazine	1 10		.....		1 10	
Spon & Chamberlain	16 50		16 50		.....	.....
Stock Second-hand *Transactions* on Hand	135 00		126 00		9 00	
Union Dime Savings Institution, New York, N. Y.—Trust Funds M. E. L. A. on deposit	1,481 23		‖1,905 15			423 92
Institution for Savings of Merchants Clerks, New York, N. Y. Reserve Fund A. S. M. E. on deposit	1,719 39		‖4,249 47			2,530 08
Institution for Savings of Merchants Clerks, New York, N. Y. Geo. W. Week's Legacy	.....		‖1,957 00			1,957 00
[illegible]	[illegible]		[illegible]		[illegible]	
Total		$88,721 16		$95,433 29		
	Liabilities.		Liabilities.			
	[illegible]		$250 62			

Institution for Savings of Merchants Clerks, New York, N. Y.						
Geo. W. Week's Legacy			$1,957 00			1,957 00
Cash, New Rivers National Bank, N. Y.			$4 68		1 07	
Total	Liabilities.	$88,721 16	Liabilities.	$95,433 29		
Reserved for Interest on Mortgage Accrued	$350 62		$350 62			
Advance Payments	197 27		162 45			34 82
Trust Funds, Mechanical Engineers' Library Association.—						
Fellowship	281 00		373 32		92 32	
Sinking	1,215 23		1,564 83		349 60	
Reserved for Uncompleted Work, Volume of *Transactions*	¶6,300 00		6,610 00		310 00	
Library Development Fund	319 39		669 84		350 45	
Reserve Fund A. S. M. E.—Life Membership	400 00		1,007 00		607 00	
" " " Initiation Fees	4,959 00		2,917 50			ϕ 2,041 50
Altoona Mechs. Library—Credit Balance with Society	19 25		18 00			1 25
Geo. W. Week's Legacy		14,041 76	1,957 00	15,630 56	1,957 00	
Surplus, September 30, 1904				$79,802 73		
" " " 1903		$74,679 40		$74,679 40		
Total Increase						$9 961 31
" Decrease					$4,837 98	$4,837 98
Net Increase in Assets = Amount written off from the liability to Reserve Fund, Initiation Fee Receipts, 1902–3, which was not covered by a Cash Deposit in Savings Bank, $2,059, plus amount of Liability to same Fund for 1903–4, written off for same reason, $5,292 = $7,351, minus Excess of Expense Incurred over Current Income 1903–4 = $2,005.17 and excess actual expenditure for completing Vol. XXIV. over amount reserved for same $222.50. That is $7,351 — ($2,005.17 + $222.50 = $2,227.67)				$5,123 33		$5.123 33

* 10 per cent. depreciation written off.
+ 10 per cent. depreciation written off, and amount expended on new furniture added.
‡ All for year 1903–4 and considered as collectable.
§ $293.90 of this sum represents a debt contracted on September 29, 1904.
‖ A total cash asset of $8,111.62, bearing interest in bank, has been created in the last two years against a total liability to funds of $8,489.49.
¶ Amount actually paid to complete Vol. XXIV. was $6,522.50—that is, amount reserved was $222.50 too small, excess being charged against surplus at September 30, 1903 (see Sheet C.).
ϕ This gain is due to the writing off of the amount credited to this fund in 1902–3 and shown as a liability, and which was not covered by a Cash Deposit in Savings Bank. *I.e.*—

Liability September 30, 1903, as shown above		$4,959 00
Less Cash Asset in Savings Bank		2,900 00
Amount written off		$2,059 00
Increased Liability—Interest 1903–4		17 50
Net gain as above		$2,041 50

REMARKS IN COMMENT ON TABULAR STATEMENT OF CHANGES IN ASSETS AND LIABILITIES.

The tabular Statement on previous page shows the financial condition at the beginning and the end of the fiscal year 1903–4, the changes in each of the accounts due to the transactions of the year, and resulting changes of Assets and Liabilities. These may be summarized as follows:

Surplus, September 30, 1903 (as reported)		$74,679 40
Less Excess Actual Cost to complete Vol. XXIV. for 1902–3, over amount reserved for that purpose, *i.e.*:		
Amount reserved at September 30, 1903	$6,300 00	
Actual Cost	6,522 50	
Excess charged against Surplus of September 30, 1903		222 50
		$74,456 90
Less Excess Expense over Income, 1903–4 (see Sheet B)		2,005 17
		$72,451 73
Add amount of liability to Reserve Fund, Initiation Fee Receipts, which is not covered by Cash Deposit in Savings Bank, 1902–3, now written off	2,059 00	
Add amount of liability to Reserve Fund, Initiation Fee Receipts for 1903–4, not covered by Cash Deposit in Savings Bank, and now written off	5,292 00	
Total Increase		$7,351 00
Surplus, September 30, 1904		$79,802 73

That is:

Amount of liability to Reserve Fund, Initiation Fees, written off	$7,351 00	
Less Excess Income over Expense, 1903–4 (Sheet B)	$2,005 17	
" Excess Actual Cost to complete Vol. XXIV. over amount reserved at September 30, 1903, charged to Surplus	222 50—	2,227 67
Net Increase in Surplus		$5,123 33

The principal items which have increased the liabilities at September 30, 1904 over those at September 30,1903 may be summarized as follows:

Geo. W. Week's Legacy Fund	$1,957 00	
Trust Funds, Mechanical Engineers' Library Association	441 92	
Reserved for completion of Volume of *Transactions*, increase this year over last	310 00	
Library Development Fund	350 45	
Credit to Reserve Fund—Life Membership	607 00	
Total Increases of Liabilities		$3,666 37
Less Decrease in Liabilities, as follows:		
Advance Payments	[illegible]	
Altoona Mechs. Library	1 25	
Amount of Liability to Reserve Fund—Initiation Fees written off	2,041 50	
		2,077 57

Less Decrease in Liabilities, as follows:				$3,066 57
Advance Payments			$24 93	
Altoona Mechs. Library			1 00	
Amount of Liability to Reserve Fund—Initiation Fees written off			2,041 50	
				2,077 57
Net Increase in Liabilities				$1,588 80
Liabilities at September 30, 1903		$14,041 76		
Add Net Increase as above		1,588 80		
Liabilities at September 30, 1904—(see Sheet C)		$15,630 56		

TRUST, RESERVE AND GEORGE W. WEEK'S LEGACY FUNDS.

Liability to Trust Funds Mechanical Engineers' Library Association (including Interest)			$1,938 15	
Cash Asset in Union Dime Savings Bank				$1,905 15
Liability to Library Development Fund, American Society of Mechanical Engineers (including Interest)			669 84	
Cash Asset in The Institution for Savings of Merchants Clerks				324 97
Liability to Reserve Funds, American Society of Mechanical Engineers—From Life Membership Receipts (including Interest)	$1,007 00			
Cash Asset in The Institution for Savings of Merchant Clerks		$1,007 00		
Liability to Reserve Funds, American Society of Mechanical Engineers—From Initiation Fee Receipts (including Interest)	2,917 50		3,924 50	
Cash Asset in The Institution for Savings of Merchants Clerks		2,917 50		3,924 50
Liability to Geo. W. Week's Legacy Fund, American Society of Mechanical Engineers			1,957 00	
Cash Asset in The Institution for Savings of Merchants Clerks				1,957 00
Total Liability to Funds			$8,489 49	
Total Asset to Funds (Cash in Savings Banks)			8,111 62	$8,111 62
Total Liability to Funds not covered by a cash asset in Savings Banks for the reason that the Income of the past two years has not allowed sufficient cash available for such purpose			$377 87	
Detail of Liability to Funds not fully covered by a cash amount—				
Trust Fund, Mechanical Engineers' Library Association, Unpaid Subscriptions considered as a Good Asset	$33 00			
Library Development Fund, A. S. M. E.	344 87			
	$377 87			

SHEET E

PRELIMINARY ESTIMATE OF RECEIPTS AND EXPENSES, FISCAL YEAR 1904–1905.

October 1, 1904, *to September* 30, 1905.

Estimated Receipts.			*Estimated Expenses.*		
Dues, 99 p. c. Gross Receipts	$38,000		House Account	$3,700	
Initiation Fees, 10 p. c. Gross Receipts	400		Interest on Mortgage	1,402	
Sales Account—Publications	3,200		Repairs and Renewals—House and Furniture	800	
House Account—Rent Sleeping Rooms	$1,200		Additions—House and Furniture	300	
" " — " Hall	60		Certificates and Membership Cards	250	
	1,260		*Transactions*—		
			Volume XXV., cost to complete	$6,610	
			" XXVI., cost for work done during 1904–5	10,000*	
				16,610	
			Circulars	2,200	
			Meetings	1,800	
			Year Books and Pocket Lists of Members	3,000	
			Salaries	10,140	
			Office Account	1,800	
			Repairs and Distribution of Badges	50	
			Library Expense Account (including Librarian's Salary)	950	
			Interest on Bank Loans		
			Committee Work—Research	250	
			Legal Expenses and Expert Fees—Legal Fees		
			" " " " " —Expert Fees	200	
			Transactions—New Edition, Vol. XXIV	740	
			Headquarters, St. Louis World's Fair	150	
			Total		$44,342
	$42,860		Library Development—New Books	$324	
Cash on Hand in Savings Bank for Library Development	$324				$44,666
	43,184				
From Reserve Fund†	$1,482				
	$44,666				

* Estimated amount to complete Vol. XXVI. that will have to be reserved at end of fiscal year, 1904-5, $5,000 additional, making estimated total cost of Vol. XXVI., complete, $15,000.

† This represents the amount which must be withheld from the annual addition to cash asset placed in Savings Bank, against the liability to reserve funds, the Council having directed that one per cent. of the receipts from dues and ten per cent. of the receipts from initiation fees be laid aside: *i.e.*, without such encroachment the reserve for 1904-5 is estimated to be $373 + $3,600 = $3,973.

PAYMENT ON LAND FOR NEW ENGINEERING BUILDING.

Cash on Hand in Savings Banks that can be applied to meet first payment—			First Payment to be met in July, 1905	$8,000	
A. S. M. E. Reserve Fund	$3,924				
M. E. L. A. Fellowship Fund	1,905				
		$5,829			
Amount to be raised to balance.....	$2,171				
		$8,000			$8,000

At the close of the printed reports, the report of the Tellers of Election was presented for record.

In presenting it the Secretary called attention to the fact that the enrollment as the result of this election was as follows:

Members and Honorary Members	1,934
Associates	236
Juniors	609
Total	2,779

The additions during the current year have been:

Members	108
Associates	38
Juniors	72

This is believed to be up to date the record for the increase in the Society membership in one year.

Messrs. John Platt, Harris Tabor, George M. Bond, Tellers for the election of officers under the provisions of the By-Laws, presented their report, as follows:

REPORT OF TELLERS.

Pursuant to the provisions of Article B34 of the By-Laws, the undersigned were appointed Tellers of Election of Officers to canvass the votes cast and report the result to the meeting.

They therefore beg to submit the following report on the ballot for officers of the Society for the year 1904-1905:

Total ballots cast	804
Ballots thrown out unsigned or otherwise informal	29
Total ballots counted by the tellers	775

As the result of their canvass, the Tellers would report the following elections:

For President.

John R. Freeman	753
Scattering	6

For Vice-Presidents.

S. M. Vauclain	748
H. H. Westinghouse	758
Fred W. Taylor	748
Scattering	1

For Managers.

Geo. M. Brill 754
Fred J. Miller 754
Richard H. Rice 751

For Treasurer.

Wm. H. Wiley 752

Respectfully submitted,

JOHN PLATT.
HARRIS TABOR.
GEO. M. BOND.

The chair, on the reading of this report, declared the members elected to their respective offices as reported by the Tellers, and asked Messrs. John Fritz and Chas. Wallace Hunt to escort Mr. John R. Freeman, President-elect, to the platform of the hall.

Mr. Hunt, as chairman of the escort committee, formally presented the President-elect to the President in office, and the latter introduced his successor to the meeting.

The President-elect then took his seat on the platform at the side of the presiding officer during part of the session.

The President then called for the docket of the professional committees now in existence, but there were no reports read at this meeting. He announced the resignation and discontinuance of the committee appointed by the Society to conduct tests on the supporting power of large-dimension I-beams, which had been proposed to coöperate with the New York Chapter of the Institute of Architects for this purpose. There had been insufficient funds available for the carrying on of the committee's work, and it had asked to be discharged, which the Council had done.

The report on a Standard Tonnage Basis for the Rating of the Refrigerating Machinery was deferred to a later session. The other committees reported progress only.

Mr. Charles Wallace Hunt, representing the appointees of the Society in connection with the work on the Union Engineering Building, reported that the houses on the site of the proposed building, Nos. 25 to 33 West Thirty-ninth Street, have been removed, and the plans for the new building to be erected thereon are substantially completed, and will be approved by the committee when the three engineering societies have passed upon

them. It is expected that the contracts will be let in the early days of January.

He called attention also to the problems before the committee with respect to organizing the three societies into one unit for the holding of the property, and how on taking the matter up with some of the best corporation lawyers in the city it was discovered that there was little or no precedent to serve as a guide. The basis of the Cooper Union seemed to be the nearest approach to the condition to be met.

This legal work has been practically completed at the present time, so that the title to the property will probably pass before the first day of January.

With respect to the work on the library to be housed in the Engineering Building, Mr. H. H. Suplee presented his report in the following form:

Mr. Suplee.—The Joint Library Committee is formed of members from each of the three societies which are to occupy the Union Engineering Building, and that committee has been actively engaged ever since its original appointment. It has held stated meetings every month and considered from every possible point that has yet been brought to them the matters necessary for the combination and continuance of the joint libraries. In addition to that they have conferred with the architects in regard to the arrangement of the floors. Among other things they have secured such modifications of the plans as will secure for the library the best possible accommodations on the upper floors of the building.

They have also had before them a matter which is not yet capable of being decided, but which I think will interest the members, and that is the nature of the work of the library as a whole. We propose to begin at first by combining the three libraries so that they will form a working library without any modification of ownership, but we hope to lay the foundation so broad and deep that the material will be forthcoming to make it the greatest scientific library in the United States.

The committee held a meeting yesterday and approved the latest plans of the architect for the two library floors in such a shape that they are now ready to be placed at the disposal of the building committee. But the committee proposes to continue its meetings until the time that the rooms are ready for occupation.

Mr. Gus C. Henning reported on the request concerning the

movement to prepare and install in the Society House a bronze bust of Prof. Robert H. Thurston, the first President of the Society, who had died in 1903. Mr. Henning's report was as follows:

Mr. Gus C. Henning.—The fund for the Thurston Memorial amounts to about $790 at the present time. It is deposited in the Colonial Trust Company to the account of Gus C. Henning, Treasurer. A model of Dr. Thurston, somewhat above life-size, has been made, but rejected because unsatisfactory. Another is now under way, and when completed will be submitted for the approval of the committee to make sure that it will be generally satisfactory.

Mr. H. H. Suplee, on behalf of the Historical Committee, consisting of Professor Sweet, Charles Wallace Hunt and Mr. Suplee, presented the following:

Mr. H. H. Suplee.—Mr. Chairman, I have only a progress report to make, but I think it will be interesting to the members, and I feel sure advantageous to the work of the committee if I do make such a report.

The committee has been engaged in collecting material to prepare a volume upon the entire history of the Society during its past twenty-five years. The earliest portion of this work was naturally the most difficult, but I am very glad to say that we have succeeded in getting together material which in a few years it would be impossible to get in any way.

Professor Sweet, the founder of the Society, has given his recollections, and has also contributed a number of interesting letters. Mr. John J. Grant has contributed a very valuable letter, being the very first of the original number of letters sent out by Professor Sweet requesting his friends to consider the formation of a Society. So we have one of the letters calling for the preliminary meeting.

We have also succeeded in obtaining a satisfactory photograph of the building at No. 96 Fulton Street, which was at that time the office of the *American Machinist*, and in which the preliminary meeting was held. Probably in a short time that would not have been possible. The sole survivor of the original firm of proprietors of the *American Machinist*, Mr. Lycurgus B. Moore, has contributed his recollections of the first meeting. So we have collected a number of very interesting records, and these, supplemented by the reports in the technical papers of that early date, will enable us to record the first early meetings very fully.

There is one point which is in the work of the committee which I think might well be brought before the members at the present time. But before proceeding to that I would like now to make a special request to all the members here, that if they have anything whatever in the form of correspondence, photographs, or anything in any way which relates to the formation of the Society it will be very desirable if they will place this matter at the disposal of the committee.

I might also add that I have secured a photograph of the room in Stevens Institute in which the first regular meeting was held.

The other matter is that of making known to the members the desire to secure historical records for the Society. We have now a number of valuable things in our present building, but we hope to get a great many more. A record of these will form a part of the historical report, and every member who has anything which is interesting in the historical mechanical line can be sure that if entrusted to the Society, either as a gift or a loan, it will be carefully preserved. These things will all be catalogued. This is as far as we have gone at present.

The Secretary in the regular order of business presented the amendment to Sections C57 and C58 of the Constitution which was presented at the Chicago meeting by Mr. F. J. Miller, and which had been sent out to the voting membership, as required by the Constitution. In a discussion on the method of procedure the President ruled that the proposed amendment to the Constitution was before the Society as a resolution, subject to a resolution for amendment, and also a resolution amending the amendment, but no further resolution could be offered until the last pending resolution was voted upon or withdrawn.

An amendment to the form and wording of the proposed amendment to the Constitution before the meeting was offered by Mr. Jacobi. The subject was discussed at length by Messrs. F. J. Miller, Jesse M. Smith, Gus C. Henning, C. W. Hunt and others, Mr. Hunt offering an amendment to Mr. Jacobi's resolution, which was subsequently withdrawn, and a vote was then taken upon Mr. Jacobi's amendment, which was lost.

A rising vote was then taken on a resolution to send the amendment to Article C57 and C58 presented at the Chicago meeting to the membership for a letter ballot. The motion was lost, but two votes being cast in favor thereof.

The Secretary read an invitation from the American Associa-

tion for the Advancement of Science and its Section D on mechanical science and engineering, asking the members of the Society to attend the meetings of that body at their December meeting in the city of Philadelphia.

Col. E. D. Meier at this point, at the request of the chair, presented the matter of securing the coöperation of the members of the Society in a movement to secure a continuance of the work which had been started by the United States Geological Survey in connection with the Louisiana Purchase Exposition for the testing of the coals of the United States with respect to their calorific powers. Colonel Meier spoke as follows:

Col. Meier.—There have been no general tests of coals available for boiler purposes in the United States since 1844. There have been some made by various railways of coals found in their tributary territory, but these were necessarily of local value and have not been made public. Now, just before the beginning of the St. Louis Exposition, some members of the Geological Survey in Washington obtained action by Congress giving them $20,000 for the purpose of testing coals available for boiler purposes from any field in the United States, the provision being (not unusual for Congressmen when they become very economical) that all apparatus must be donated. This made a great deal of difficulty, but they succeeded in getting together quite a respectable little plant. They have three boilers, a gas-producing plant and a gas engine and a sufficient chemical laboratory. By the coöperation of some gentlemen connected with several Western technical schools, they also obtained the services of a number of competent persons to make these tests. The plant is located in what is known as the Mining Gulch of the St. Louis Exposition, and owing to the circumstances related it was not until, I think, September before they could begin to make any tests. The manner in which the tests were conducted was this: they notified coal miners by the press and other means that they could have their coal tested there gratis, provided they furnished at least a carload without cost, they paying the freight to the Exposition grounds, and, furthermore, that each carload of coal must be first submitted to an expert of this commission to judge that it fairly represented the average output of the mine. In other words, the commission sent an expert to the mine and selected a standard carload of coal just as it was mined. The commission has succeeded, but the last I heard from there was that they had made

some eighty-five tests and wanted to make a great many more. They have about one hundred and fifty sample carloads of coal offered them now. They cannot accept them unless there is some provision made for the cost of continuing the tests. The Fair officials and city officials of St. Louis and others have consented to allow that plant to remain there for some six months longer, and most, if not all, of those who contributed to the plant have agreed to leave their apparatus that much longer.

The commission estimates that in order to do this thing thoroughly it should be made a much more permanent plant than it is, and they are going before Congress, or have gone before Congress, through Mr. Hemingway, asking for an appropriation of $100,000, and if this is passed they will continue the tests at once, until all who have applied can have their coal tested. They hope to have by that time such an amount of valuable information that Congress may be induced to have this testing plant made permanent and very much enlarged; the site where it will then be placed has not been selected.

At present they wish to continue, and if the resolution passes they may have some 250 sample carloads to test. Now, all those who have to do with steam-boiler problems in the West know that there is very little known about Western coals. It is just as important to the Eastern as to the Western engineer to have these tests made, because consulting engineers and contracting engineers here are asked to make promises or even guarantees in regard to the performance of certain plants on local coals in all sections of the United States.

Now, the St. Louis Engineers' Club, which is the local engineering society, has passed a resolution which I would like to read to you. It was adopted on November 16, 1904:

That most active member of our Society, Mr. Bryan, of St. Louis, has written to Professor Hutton, asking for some action by the Society or Council, and Professor Hutton, while very much interested in the project, has answered that neither the Council nor the Society is very much enamored of taking anything to Congress which may look like lobbying anything through for the benefit of the Society. I would like, however, to have the individual members take an interest in this by either interviewing or writing to their Congressmen or Senators to show that the engineering profession as a whole is interested in the continuance of these tests, and in order to bring that about I would like to have

you pass a resolution instructing the Secretary to present this matter in a short circular to the membership at large, asking them to write to the Congressmen of their own districts to get them to assist the authorities of the Geological Survey.

Resolved, That it is the sense of this meeting that the Secretary be instructed to set forth in a short circular the facts of this coal testing plant, the desirability of its continuance and to suggest to the individual members of the Society that they use their influence with their congressmen and senators in having the bill passed, authorizing the continuance and providing the necessary funds.

The exact form of this resolution, however, to be left to the President and Secretary.

After brief comment with respect to the quality of the work done by the plant, Colonel Meier's resolution was carried. Professional papers were then taken up as follows:

"A New Hydraulic Experiment," by A. F. Nagle; "A Twist Drill Dynamometer," by W. W. Bird and H. P. Fairfield; "Diamond Tools," by Gus C. Henning.

The participants in debate were G. W. Colles, R. C. Carpenter, Benjamin Baker, H. H. Suplee, A. C. Walworth, H. Emerson, F. W. Taylor, S. W. Baldwin and W. J. Kaup.

Wednesday afternon was left free for the convenience of members attending to personal and professional business in the city.

Third Session. Wednesday Evening, December 7th.

The meeting was called to order at 8:30 in the Hall of the Mendelssohn Union. Professional papers were presented as follows:

"Centrifugal Fans," by A. J. Bowie, Jr.; "Computation of Values of Water Powers, and Damages Caused by Diversion of Water Used for Power," by Charles T. Main; "An Indicating Steam Meter," by Charles E. Sargent; "Stay Bolts, Braces and Flat Surfaces: Rules and Formulæ," by Robert S. Hale; "Condensers for Steam Turbines," by George I. Rockwood.

The participants in debate were R. C. Carpenter, G. C. Henning, H. H. Suplee, W. T. Magruder, G. R. Henderson, W. S. Huson, F. B. Perry, Joseph Morgan, F. Hodgkinson, G. J. Foran, F. M. Wheeler and W. E. Crane.

In view of the curtailing of the discussion in the morning for lack of time, Mr. Henning was requested by the chair to contribute further to the discussion on the cutting properties of the diamond tools.

Fourth Session. Thursday Morning, December 8th.

Professional papers were taken up as follows:

"Bursting of Four-Foot Fly-Wheels," by Charles H. Benjamin; "Influence of Connecting Rod upon Engine Forces," by Sanford A. Moss; "Losses in Non-Condensing Engines," by James B. Stanwood; "Power Plant of Tall Office Buildings," by Sterling H. Bunnell; "Pressures and Temperatures in Free Expansion," by A. Borsody and R. C. Cairncross (presented by Charles E. Lucke).

The participants in debate were H. H. Suplee, G. C. Henning, G. R. Stetson, J. E. Johnson, Jr., F. A. Halsey, G. R. Henderson, F. H. Ball, J. B. Stanwood, R. P. Bolton and F. R. Hutton.

Fifth Session. Friday Morning, December 9th, 1904.

This session was called to order in the auditorium of the Society's House, No. 12 West Thirty-first Street, at 10 o'clock. The President on taking the chair announced to the meeting that at the last meeting of the Council Mr. Thomas A. Edison, who had been a life-member of the Society from a very early date in its history, had been elected by the Council to honorary membership, and had signified his acceptance of the honor. This announcement was received with applause. Professional papers of the morning were then taken up as follows:

"Fuel Consumption of Locomotives," by George R. Henderson; "Road Tests of Brooks' Passenger Locomotives," by E. A. Hitchcock; "Discharge of Water with Steam from Water Tube Boilers," by A. Bement; "More Exact Method for Determining the Efficiency of Steam Generating Apparatus," by A. Bement; "Forcing Capacity of Fire Tube Boilers," by Francis W. Dean.

The participants in debate were R. C. Carpenter, E. A. Hitchcock, W. F. M. Goss, H. Emerson, C. B. Rearick, A. A. Cary, O. H. Woolson, W. O. Webber, G. I. Rockwood, William Kent, H. deB. Parsons, G. R. Henderson, A. Bement, H. H. Suplee, A. Herschmann, G. H. Barnes, D. S. Jacobus, G. I. Bouton, G. M. Brill, R. H. Rice, J. J. Hoppes, A. H. Eldredge and Abercrombie.

At the close of the papers the committee appointed to suggest a standard tonnage basis for refrigerating machinery presented, through Prof. D. S. Jacobus, chairman of the committee, this pre-

liminary report, which is printed as a paper contributed at this meeting.

Mr. Kent called attention to the recommendation that the committee should be asked to suggest rules for the best method of measuring the quantity of fluid evaporated in refrigerating, together with a code of rules of conducting tests of refrigerating machines or plants.

On motion, the committee was requested by the meeting to proceed with the carrying out of this recommendation.

At a meeting of the Society held in New York December, 1901, a committee reported on the standardization of the engine and dynamo for electric generating plants when both pieces of apparatus were not to be furnished by the same builder. This report had received the usual treatment given such contributions, and the committee had been discharged from further consideration of the subject.

The practical use of the report in the hands of those interested had developed the wisdom of incorporating a further provision, and the individual members of the Society who had constituted the original committee united in presenting and seconding a resolution that an additional heading be added to the report. (See Paper No. 1056.) The resolution presented and seconded by these members was as follows:

Resolved, That the recommendation of Messrs. J. B. Stanwood, W. M. McFarland, A. L. Rohrer, Frank Ball and W. B. Forbes be treated as an appendix to the original report, and printed for circulation with that report, and to members of the Society, as the report shall be requested for future distribution.

On motion, the resolution was adopted and its recommendations approved.

The chair called the attention of the members to the presence in the auditorium of a bronze bust of the late Capt. John Ericsson, who had been during his lifetime so distinguished a member of the Society. He called on Mr. Dodge to present for record certain matters of historical interest in connection with the bust, as follows:

Mr. J. M. Dodge.—The bust is the work of Prof. John Kneeland, who was a personal friend of Captain Ericsson. Captain Ericsson was a recluse during nearly all the time he was in this country. My grandfather, Prof. James J. Mapes, was perhaps his most intimate friend. While Ericsson was working around his

shop he allowed Professor Kneeland to make the bust. Ericsson was very much pleased with it. There was, I believe, only one plaster cast made, and this was presented to the Society by my uncle, Charles Mapes. To insure permanency it has been cast in bronze. When I was a little boy my earliest recollection was seeing the plaster bust, which was here until a few days ago, in our house. I remember as a child it was Captain Ericsson to me. So it is a rather unique treasure, and I am sure as time goes on it will be more and more appreciated. You can see the resemblance to his picture on the wall of this auditorium. This picture was made when he was a very much older man. At that time, as I remember, or believe, Ericsson was about thirty-five. He approved of it himself, and so also did his friends.

The President then called on Mr. John R. Freeman, President-elect, to step to the platform, and turned over the office of President to his successor. Mr. Freeman spoke of his appreciation of the work that had been done by the President, Council and other officers of the Society in arranging and administering the details of the Society's affairs, outside of the public meetings, and hoped that he would be able to maintain the standard of his illustrious predecessors.

The meeting was then adjourned.

On the afternoon of Thursday the members were the guests by invitation of the engineers of the Rapid Transit Commission at the power-house of the subway at Fifty-ninth Street and the North River.

Parties were made up in small groups under guides from the local membership and escorted to the power-house through the subway, which had been in operation only since the 27th of October. The visit was much enjoyed.

On the evening of Thursday the social reception given by the local membership to the visiting members was held in the ballroom of Sherry's, Forty-fourth Street and Fifth Avenue. Over six hundred members and ladies were in attendance.

On Wednesday and Thursday luncheons were served between twelve and two in the auditorium of the Society House, which had been cleared for this purpose.

The spring meeting of the Society is to be expected in the city of Scranton, Pa.

No. 1052.*

SOME REFINEMENTS OF MECHANICAL SCIENCE.

BY AMBROSE SWASEY, CLEVELAND, O.

PRESIDENT'S ADDRESS, 1904.

As we open this, the Twenty-fifth Annual Meeting of the American Society of Mechanical Engineers, the history of the Society for a quarter of a century comes before us, and it is an occasion when it is especially appropriate to make some mention of the growth and progress of the Society since it was organized.

At the beginning of the Society, who would have dared to predict the wonderful advance that has been made in mechanical engineering. There was indeed a great field for work for just such a Society. The long list of meetings which have been so fully attended, and so valuable to the members; the transactions, with their records of addresses, papers and discussions by men of experience in nearly every branch of mechanical engineering, and the constant growth of the Society until at the present time it has a membership of nearly twenty-nine hundred, all go to show that from the beginning it has been an earnest and progressive organization, and a most important factor in the progress of mechanical science and of the mechanic arts.

Not only those of us who were counted among its first members but those who from year to year have been added to its membership, may well feel proud of its splendid record.

The scope and influence of the Society, which has been constantly increasing in the past, will surely continue, and never was its future brighter than at present.

For the subject of my address, I wish to speak of a few of those methods and mechanisms which have been developed and perfected to such a degree of refinement that they may be con-

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

sidered as almost beyond the practical, and yet were it not for such refinements they could not possibly be made to serve the utilitarian purposes which make them of such inestimable value to us all.

The division and the measurement of time is to-day, as it has been for ages, among the most important of the subjects affecting the welfare of mankind, and as time has rolled on and there has been a better understanding of the laws governing the universe, nearer and nearer has been the approach to perfection in the working out of these difficult problems, but the many limitations surrounding them have always kept their full solution somewhere in the future.

The diurnal revolution of the earth, which gives the solar day, and the revolution of the earth around the sun, the solar year, are the arbitrary divisions of time marked off with the utmost precision by the celestial bodies; and while the length of the solar day has, from before the Christian Era, been fairly well defined, the length of the solar year was but approximately known until within a few hundred years.

The length of the year as counted by the Julian calendar was too long by eleven minutes and fourteen seconds, and this error amounted to ten full days in the sixteen hundred years from the time the Julian calendar went into effect, until the introduction of the Gregorian calendar.

A few years ago, when visiting the Vatican Observatory, I was particularly interested in the Gregorian Tower, which forms a part of the Vatican Library Building. After passing through a number of rooms which are used in connection with the Observatory, when near the top of the tower, I was taken into the spacious and beautiful calendar room, the walls of which are covered with paintings of the highest order, executed centuries ago, under the direction of Pope Gregory XIII. In the center of the room, and forming a part of the floor, there was a large marble slab, on which was cut a fine line exactly in the true meridian, and upon the line was a special mark which indicated the altitude of the sun at noon of a certain day. On the south wall, near the top of the room, there was a small aperture through which the direct rays of the sun passed at noon, projecting a bright spot on the meridian line.

All of this had been planned and executed by the astronomers in order that they might demonstrate the necessity of reforming

the calendar, and when at noon on the 11th of March, 1582, Pope Gregory saw that the altitude of the sun as shown by the beam of light, was not for that particular day, but for the day ten days later, he directed that ten days be stricken from the calendar, and that day should be the 21st of March, instead of the 11th.

With such precision had the astronomers determined the true length of the year, that our present calendar with its intercalations will continue on for twenty thousand years, with an error not to exceed a single day.

The line on the marble slab and the aperture through the wall of the Calendar Room were devices simple in the extreme, and in this day of instruments such a method would hardly be considered, yet they served their purposes admirably, and the placing of that line on the true meridian, with an accuracy never before attained, was considered one of the greatest scientific achievements of that age.

Since an unknown time the day has been divided into twenty-four hours and as civilization has advanced, the greater has been the necessity for the utmost precision in the measurement of each hour, with its subdivisions.

The sun dial is not only the earliest, but the most interesting of all the numerous arrangements that have been devised for measuring the divisions of the day. Notwithstanding its limitations, it has been a subject which has attracted the brightest minds for ages. Within these later years there has been a renewed interest in this ancient time-keeper, not only in copying the types of dials, which are valuable because of their antiquity, but in working out new forms. Recently a new dial has been invented, by which the rays of the sun will indicate the true mean time for each day of the year, with an error not to exceed one minute.

The hour glass, which came later, was considered a much more practical method, inasmuch as it could be used either day or night, and because its use was not confined to a particular location; however, as a time-keeper it was not satisfactory, even in those early days.

The clepsydra or water clock, which is supposed to have been invented by the Greeks, was found to be a much better time-keeper than either the sun dial or hour glass, and it was a great step in advance toward the accurate measurement of time.

These water clocks are to this day used extensively in the East, more especially in China. Those first used by the Greeks consisted of two water jars so arranged that the water from the upper ran into the lower, and the time of day was determined by measuring the depth of water in the upper jar, and at sunrise each day the water was returned to the upper jar. In the City of Canton there is a water clock which has been running for eight hundred years, and at the present time it is the standard clock of that city.

This clock consists of four water jars, each having a capacity of eight or ten gallons. The jars are placed one above the other, in the form of a terrace, the three upper ones being provided with a small orifice near the bottom, through which the water drops into the jar next below, and so on down from one to the other, until the water reaches the lowest or registering jar. In this there is a float, to which is attached an upright, having graduations for the hours and parts of hours, and as the water rises the time can be determined by noting the height of the float in relation to the cross-bar at the top of the jar.

In this improved form of water clock the variation in the flow of water due to the difference in height, is overcome by having a series of jars, the outlet of the upper being so graduated that there is but little variation in the height of water in the second jar, and in the third the height remains practically uniform, thus insuring a constant head for the water which drops into the registering jar. At the beginning of each day the water is taken from below and carried up a flight of steps to the top.

That such an arrangement has some elements favorable to the accurate measurement of time, there can be no doubt. It certainly has the element of simplicity, and notwithstanding its long service, the only wear noticeable, was confined to the steps leading to the upper jar.

Clocks of the present type, although used as far back as the twelfth century and possibly earlier, were but fair time-keepers, until several centuries later. Those which the astronomers used in their observatories at the end of the fifteenth century, were so unreliable that modified forms of the clepsydras of the ancients were used, and, as they did not prove to be satisfactory, most of the observations were made without the use of clocks.

Galileo's beautiful discovery of the isochronism of the pendulum from the swinging chandelier in the church at Pisa, was of

great value in many respects, but in none more so than in its application to the measurement of time.

Soon after that great discovery the English clock-maker, Graham, invented the mercurial pendulum, by which the variation in its length caused by the difference in temperature was fully compensated, and some years later Harrison, another English clock-maker, invented a compensating pendulum, which consisted of a series of metal bars having different coefficients of expansion—so that two hundred years ago, as it is to-day, the pendulum was the nearest perfect of all the devices that have been employed for governing or controlling the motions of a clock mechanism.

Every part of the clock down to the minutest detail has been the subject of study and improvement, and they are made and adjusted with such precision and delicacy that in testing them the question is, within how small a fraction of a second will they run. Not content with their marvelous performance when under normal conditions, some of the finest astronomical clocks are surrounded by glass or metal cases, in which a partial vacuum is maintained, and in order that the cases may not be opened or disturbed, the winding is done automatically by means of electricity; the frequency of the winding in some cases being as often as once every minute. These clocks are set up in especially constructed rooms or under-ground vaults, where they are free from jar or vibration, where the temperature and barometric conditions remain practically constant, and where every possible precaution is taken to further minimize the errors of the running rate.

A clock in the Observatory at Berlin has run for several months under these favorable conditions, with a rate having a mean error of but fifteen one-thousandths of a second per day and a maximum error of thirty one-thousandths of a second per day.

Another clock installed at the Observatory of Case School of Applied Science at Cleveland, running under similar conditions, also has a mean error of fifteen one-thousandths of a second per day, with a maximum error for several months of but twenty-two one-thousandths of a second per day.

These are notable examples of the present state of the art of clock-making, and show the wonderful precision with which minute intervals of time can be measured.

From the time of the invention of Peter Hele, in 1477, of the "Nuremburg Animated Egg" or "pocket clock," which required

winding twice a day, and varied an hour and a half in the same length of time, the development of the watch has kept pace with the "Mother Clock," and followed closely to it in time-keeping qualities.

These marvelous little machines, whether made at the homes of the peasants among the hills and mountains of Switzerland, where the skill required for making a single part has been handed down from generation to generation, or made in the great factories of this country, where fully two million high-grade movements are turned out annually, and where the skill of the workmen has been supplemented by modern methods and machinery, are, notwithstanding the difficulties attending their manufacture, produced so cheaply as to be within the reach of almost every one.

The larger watch, or ship chronometer, with its escapement so delicately made and adjusted that it must always be kept in the same position, was greatly improved through the efforts of the British Government in 1714 by offering rewards of ten, fifteen and twenty thousand pounds to any who should make chronometers that would run so accurately that the longitude of a ship at sea could be determined within sixty, forty and thirty miles, and Harrison, the inventor of the compensating pendulum and the compensating balance, which is now used in watches, succeeded in making a chronometer, which, after being tested on a long voyage, was found to run so closely that the position of the ship was determined within eighteen miles, and he was therefore paid the full award of twenty thousand pounds. That historic chronometer, which marked a new era in navigation, is now numbered among the treasures of the Greenwich Observatory.

Modern ships are equipped with chronometers so accurate and so reliable, and with sextants of such precision that navigators can determine their position in latitude and longitude within a few miles. Therefore, with the increased speed of the powerful ships carrying hundreds or even thousands of passengers, together with their valuable cargoes, the methods and instruments used in navigation have been so improved as to greatly diminish the dangers in crossing the seas.

The perfection attained in the measurement of time, which is of such great practical value in nearly every sphere of life, would not have been possible were it not for the even greater refinements that have characterized the methods and instruments used by the astronomer in determining the length of the day and of the year, which are the fundamental standards of time.

The division of the circle and the measurement of angles has ever been among the unsolved problems of the astronomer, yet in the instruments used by him, circles have formed a most important part.

Long before the telescope was invented, Tycho Brahe, the Danish astronomer, "the founder of Modern Astronomy," constructed for his observatory, instruments of various kinds having graduated circles and arcs of circles. His instruments for the most part were improvements on those used by Arabian astronomers in the eighth and ninth centuries, and these, in turn, were copied after similar instruments used by the Greeks and Egyptians, a thousand years previous, and it is supposed that such instruments were used by the Chinese at an even earlier period, so that graduated circles have come down to us from the far-off ages.

The longer the radius the more accurate the graduations, was the principle upon which the early instruments were made. The Arabians, in about the year 1000, built a sextant with a sixty foot radius and a quadrant with a twenty-one foot radius, but to Tycho Brahe is due the credit of constructing instruments having circles, much smaller in diameter and graduated with a greater precision than ever before. It was by the use of such improved instruments of his own making, and by his observations which were made without a telescope or any means of magnification, that he was able to give the positions of a large number of stars within less than one minute of arc from the positions given by modern astronomers.

The graduation of an eight foot mural circle in 1725 by Graham of England, for the National Observatory, and of an eight foot quadrant by Bird, in 1767, were notable steps in advance in the division of the circle and the measurement of angles, but these and similar instruments, although their efficiency was greatly augmented by the use of the telescope, have been supplanted by others more practical.

The first circular dividing engine was made in 1740 by Henry Hindley, of York, England, for cutting the teeth of clock wheels, and it is interesting to note that in the same year Huntsmann, another clock-maker, of Sheffield, invented the process of making crucible steel, that he might have a metal suitable for the springs of his clocks.

Of the several engines constructed later, the one most suc-

cessful and representing the greatest progress, was that made by Ramsden, in 1777. This engine, automatic in its movements, was made especially for graduating circles, and because of the great precision with which he divided the circles of the instruments used by the Government, the Board of Longitude awarded him the sum of six hundred and fifteen pounds. A further and most potent recognition of the excellence of his work lies in the fact that all subsequent circular dividing engines have followed closely the same general principles of construction embodied in the Ramsden engine.

It is most gratifying to all those who are interested in mechanical progress that the Ramsden engine has been preserved throughout all these years, and it now stands in the Museum of the Smithsonian Institution, at Washington, as a monument to the one who made it, and as the best example of that time of the art of graduating circles.

Many excellent dividing engines have been made that are quite sufficient in point of accuracy for the work for which they were intended, but the perfection required in the graduation of circles for astronomical instruments, is such that it has been found to be one of the most difficult of all mechanical problems to make an engine that will meet such requirements.

In such an engine the chief essential is that the spindle carrying the master-plate shall be as nearly round and as closely fitted in its bearings as is possible, for the degree of excellence with which that work is done determines how closely a circle can be divided.

It seems almost incredible that a well-lubricated spindle of four inches in diameter at its largest part and tapering three-quarters of an inch to the foot, can be made so nearly round and so closely fitted in its bearings that a movement of one-thousandth of an inch in or out of its bearings, will in one case cause it to turn with difficulty, and in the other with perfect freedom, yet this has been found to be within the limits of mechanical refinements.

The greatest accuracy thus far attained in such engines, is one second of arc, which arc, with a radius of three miles equals one inch, and at twenty inches, which is the radius of the silver ring upon which the graduations on the master-plate are made, a line one-thousandth of an inch in width is equal to twelve seconds of arc, or twelve times the accumulated errors of any number

of divisions, or twenty times the greatest error of any single division.

In automatically graduating a circle, it has been found to be impracticable to cut more than six lines in a minute, and it requires about thirty-three hours to divide a circle into two-minute spaces. As with the running of the finest clocks, the best results can only be obtained when the engine is surrounded with every favorable condition possible. Instead of the large circles and sectors used by the ancients, they have been made smaller in diameter as the methods for graduating have been improved, until those of the more modern instruments are seldom more than thirty inches, and some of the latest meridian instruments have circles of but twenty-five inches.

The smaller circles, which can be made and graduated with greater precision than the larger ones, are also less liable to change in form, owing to their weight and the variation in temperature, and with the aid of the reading microscope the results obtained would not be possible with the larger circles.

A twenty-five inch circle read with a microscope having a power of forty, would be equivalent to a circle of about eighty feet in diameter, and a single second of arc as seen through the microscope would be equal to 0.0024 of an inch, a quantity easily subdivided.

A most important adjunct to the astronomer's instrumental equipment, is the filar micrometer. With it he determines the errors of divisions, the eccentricity of his circle and measures the angles to within a fraction of a second; and when used at the eye end of the telescope he determines the positions and motions of the stars and the distances and diameters of the planets. In these little instruments, whether of the simple or complex form, the chief requisites are the screw and the cross wires, for upon them the value of the observations and measurements depend.

To make the screw of a micrometer so true that the errors in the threads cannot be detected by its own magnifying power, is an extremely difficult task. These micrometer screws are often made with one hundred threads to the inch, and are provided with graduated drums having one hundred divisions, the readings being made in tenths of a division.

The cross wires, which are but common spider lines, because of their fineness and the remarkable qualities they possess, are indispensable in micrometric work.

That the repulsive and even dangerous spider has plenty of enemies among the human family, there can be no doubt, yet if the value of the contributions which it has made to the cause of science was generally known, it would surely have a greater number of friends than at present, and most certainly the astronomer will say naught against it, for after the experience of many years he has found that the spider furnishes the only thread which can be successfully used in carrying on his work.

The spider lines mostly used are from one-fifth to one-seventh of a thousandth of an inch in diameter, and in addition to their strength and elasticity, they have the peculiar property of withstanding great changes of temperature, and often when measuring the sun spots although the heat is so intense as to crack the lenses of the micrometer eye-piece, yet the spider lines are not in the least injured.

The threads of the silkworm, although of great value as a commercial product, are so coarse and rough compared with the silk of the spider that they cannot be used in such instruments.

Platinum wires are made sufficiently fine, and make most excellent cross wires for instruments where low magnifying powers are used, yet as the power increases they become rough and imperfect.

Spider lines, although of but a fraction of a thousandth of an inch in diameter, are made up of several thousands of microscopic streams of fluid, which unite and form a single line, and it is because of this that they remain true and round under the highest magnifying power.

An instance of the durability of the spider lines is found at the Allegheny Observatory, where the same set of lines in the micrometer of the transit instrument has been in use since 1859.

The placing of the spider lines in the micrometer is a work of great delicacy, and in some micrometers there are as many as thirty which form a reticule, with lines two one-thousandths of an inch apart, and parallel with each other, under the highest magnifying power.

Step by step from the methods of the Arabian astronomers to the time of Tycho Brahe and on down to the present day, improvements in the instruments and methods for the measurement of angles have been going on, until astronomers can measure double stars with a separation of one second of arc, and within less than one second they can define their positions in the heavens.

In the realm of the measurements of minute linear distances, and the perfection of curved and flat surfaces, the refinements are even greater than those pertaining to the measurement of time and of angles.

Most important in the linear dividing engine is the screw, and although much had been accomplished in bringing such engines to a high degree of excellence, it was for Prof. Rowland to make an engine which has a practically perfect screw; and without doubt it is in all respects the nearest perfect of all the mechanisms that have been employed for ruling lines exactly parallel and equally spaced.

The Rowland engine was made especially for ruling diffraction gratings which are made of speculum metal, and with it a metal surface has been ruled with 160,000 lines, there being about 29,000 to the inch, and as many as 43,000 lines to the inch have been ruled.

The gratings mostly used have from 14,000 to 20,000 lines to the inch, and with such exactness is the cutting tool moved by the screw that the greatest error in the ruling does not exceed one millionth of an inch.

The production of these gratings, which has enabled the physicist in his study of the spectrum to enter fields of research before unknown, has not only called for the highest degree of perfection ever attained in the spacing of linear distances, but it has also called for a refinement most difficult in the optical surfaces upon which the lines are ruled. To Mr. Brashear was given the problem of producing such surfaces, and notwithstanding the many difficulties encountered in working and refining the speculum metal plates, he has made many hundred plates with surfaces either flat or curved with an error not to exceed one-tenth of a wave length of light, or one four hundred thousandth of an inch.

As the established standards of length which are the yard of Great Britain and the meter of France, are made of metal, and liable to destruction or damage, Prof. Michelson conceived the idea of determining the lengths of these standards in wave lengths of light, which would be a basis of value unalterable and indestructible.

For the purpose of carrying out these experiments, the interferometer was constructed, an instrument which required the highest order of workmanship and the greatest skill of the optician. Again Mr. Brashear proved to be equal to the occa-

sion, and made for the instrument a series of refracting plates, the surfaces of which were flat within one-twentieth of a wave length of light, with sides parallel within one second. This was the most difficult work ever attempted in the refinement of optical surfaces.

Professors Michelson and Morley devised a method for using the interferometer to make the wave length of some definite light an actual and practical standard of length. So satisfactory was the result that Prof. Michelson was invited to continue the experiments at the Bureau of Weights and Measures, at Sèvres, France, where the standard meter, which is kept in an underground vault and inspected only at long intervals, was used for that important work. The final result of the experiments, which occupied nearly a year, shows that there are 1,553,164.5 wave lengths of red cadmium light in the French standard meter, at 15 degrees Centigrade. So great is the accuracy of these experiments, that they can be repeated within one part in two millions. Should the material standard of length be damaged or destroyed, the standard wave length of light will remain unaltered, as a basis from which an exact duplicate of the original standard can be made. These two marvelous instruments, the Rowland dividing engine and the Michelson interferometer, show the possibilities in the perfection of linear divisions and the standards of length.

We have recounted some steps of the progress that has been made in the measurement of time, of angles, and of length, together with some of the refinements in these measurements, but we are confronted with the fact that notwithstanding all that has been accomplished from centuries past down to the present time, there are, as ever before, many imperfections requiring new problems in mechanical science to be worked out for the further enlightenment and welfare of mankind.

No. 1053.*

ADDRESS OF MR. CHARLES WALLACE HUNT

ON THE

UNVEILING OF A PORTRAIT OF PAST-PRESIDENT JOHN E. SWEET, DECEMBER 6, 1904.

The prominent position now held by engineers and by the profession of engineering is of a comparatively recent date. The national engineering societies that have grown up in one generation have become important and influential institutions of a semi-public character. The rapid growth of the American Society of Mechanical Engineers in membership and in influence is an interesting example of the benefits arising from the professional association of engineers, as distinguished from the results of isolated work and duplicated experiments. The scientific societies have supplied the needed facilities for the members to discuss their investigations, compare their experiences, and to place their data on record, where they are accessible to any engineer for utilization in any part of the world. The social influences naturally following the frequent meeting face to face of earnest men, swept away a host of fancied differences, leading the way step by step to mutual helpfulness and co-operation.

The influential position now held by this Society, came not by happy accident, but by the earnest work of men who stamped their own character upon it, for this Society, like all human institutions, is what its members have made it. If it has high ideals it is because its members cherished such ideals in their thoughts; they hoped for their realization; and, finally, by well directed efforts, saw them crystallize in this institution.

Twenty-five years ago Mechanical Engineers were scattered

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

throughout our country, designing and manufacturing machinery, with inadequate means of comparing their own ideas with those of their brethren, little or no co-operation with each other, and with no social bond connecting them. This isolation of those who should be companions was keenly felt by many, but especially by Prof. John E. Sweet. In 1879, he on his own initiative wrote to the correspondents of the technical newspaper, known as the "American Machinist," asking them to assemble in New York for a double purpose; to pay their respects to the editor of the paper and for mutual acquaintance. The result of this gathering was inspiring and there soon followed a call for a meeting to form a permanent organization signed by Prof. John E. Sweet, Alexander L. Holley, and Prof. R. H. Thurston. On February 16, 1880, at 96 Fulton Street, in New York City, about forty engineers met in response to the invitation. Prof. Sweet called the meeting to order, Alexander L. Holley was chosen Chairman, a name for the Society was selected, by-laws submitted, after which the meeting adjourned to April 7th, when the organization was completed. The first professional work of the Society was the passage at this meeting of a special resolution inviting Prof. Sweet, in view of the eminent services rendered by him in the organization, to read the first paper before the Society. Thus the "American Society of Mechanical Engineers" started on its beneficent and useful course, to be devoted, as Chairman Holley expressed it, "to mechanics and dynamics."

As the years passed by, the Society gained in membership, in technical standing, and in influence in the manufacturing world. With prosperity many contributions and gifts of historical material came to the Society, some of inestimable value, among which were portraits of men eminent in the advancement of engineering theory or practice. It is a pleasure and an inspiration to view the portraits that adorn the walls of our Society headquarters and contemplate the lives and the works of the eminent men who are there grouped together. Let us for a moment glance at that goodly array of those whom every one classes as prominent among the world's benefactors.

There on canvas is James Watt, the discoverer of cylinder condensation; the economy of steam expansion; who first introduced the steam jacket; the double acting steam cylinder; the application of steam to rotating shafts; the parallel motion for piston rods; the centrifugal governor; the throttle valve; the steam

pressure gauge; the water level gauge; and the steam indicator. Prof. Thurston has aptly characterized him as a "genius of the brightest intellect, a mechanic such as the world rarely produces."

Nearby is a portrait of John Ericsson, one of the greatest, if not the greatest mechanical engineer that any age has produced. While living he was the recipient of numerous diplomas, medals, titles, and orders of knighthood, and on his death his remains, with the highest honors, were returned in a United States naval vessel to Sweden, his native land. This portrait is especially prized as being the only one ever painted from life. He lived in such personal seclusion that he saw little of his fellow workers. This wonderful man, whose features were so little seen when living, all can now contemplate at their leisure. Purely by engineering ability did he twice revolutionize the navies of the world. His contributions to the mechanical world were so numerous and important that his work in engineering is in some degree similar in scope to Shakespeare's contribution to English literature.

We contemplate with especial interest the portrait of Robert Fulton, painted by himself, for he was a portrait painter before he surprised the world with the successful application of steam in navigation. It was an active and a fruitful life that he led, from the successful "Clermont" ascending the Hudson River against the tide, to the United States steamer "Fulton," the first steam war vessel ever constructed. Admirers of him have placed this portrait in our gallery, where it may have a permanent home with congenial surroundings. The bronze bas-relief on the monument erected by this Society to mark his last resting place in Trinity Churchyard was modeled directly from this portrait.

Sir Henry Bessemer now attracts our attention, and we are startled by attempting to imagine the condition of the world, without the revolution in steel making that this epoch maker developed.

Over the platform hangs the portrait of Alexander L. Holley, charming in personality and famous for the skilful introduction of the Bessemer process of steel making in this country. He signed the call for the organization of our Society and was its first Chairman.

Kindly and genial John Fritz here bears the contented expression that we expect to see in one who has earned and won the famous Bessemer medal. His great achievements in industrial affairs are emphasized by the John Fritz prize medal that here-

after will yearly be an incentive to important and skilful engineering work.

Bessemer, Holley, Fritz—what wonderful materials and processes this great trio has revealed to us!

Nearby is J. F. Holloway, of pleasant memory, one to whom so many members of the Society were indebted for a kind word or a helpful suggestion. His descriptions of early engineering experiences to be found in our transactions, are literary gems as well as an accurate record of early engineering events.

The time is too short to more than refer to the portraits of Rankine, a genius in analytical mechanics, of Dr. Francis Reuleaux, so distinguished in theoretical and applied kinematics, of George H. Corliss, of H. R. Worthington, of John C. Hoadley, of Eckley B. Coxe, of James Dredge, and of Joseph Harrison, Jr. The mention of their names brings a multitude of memories of their personality, their struggles, and their triumphs. For struggles in abundance they had, as no man ever worked himself anywhere in a dead calm.

Each one in this gallery by his engineering ability contributed in a marked degree to the advancement of science and the mechanic arts, thus making the life of each one of us now easier and more productive, as well as fuller of intellectual pleasures. The world is better and broader because they lived in it. Will not this gallery be, as it were, a helping hand to countless engineers who will come after us? Many a true heart will be given renewed courage, many a struggle will seem lighter, and the successes of life will seem dearer from the contemplation of those who blazed the way to broader engineering fields and greater business opportunities.

In that gallery of eminent engineers there is now to be placed the portrait of a man who, measured by the highest standards of honorable manhood, professional ability and earnest endeavor, will be rightly installed in that illustrious company. In behalf of the generous donor, I present to the American Society of Mechanical Engineers, a portrait of our beloved Founder—Prof. John E. Sweet.

No. 1054.*

PRELIMINARY REPORT OF A COMMITTEE TO CO-OPERATE IN STANDARDIZING ABBREVIATIONS, SYMBOLS, PUNCTUATION, ETC., IN TECHNICAL PAPERS.

This Committee is the result of a desire of the authorities in charge of the publications of the four national engineering societies to co-operate in this matter.

The members of the Committee are the following:

CHARLES WARREN HUNT, Secretary of the American Society of Civil Engineers.

D. S. JACOBUS, Vice-President of the American Society of Mechanical Engineers.

JOSEPH STRUTHERS, Assistant Editor of the American Institute of Mining Engineers.

CARY T. HUTCHINSON, Chairman of the Editing Committee of the American Institute of Electrical Engineers.

This Committee has held several meetings; it seemed advisable, at the outset, to limit its discussions closely to the general subject of abbreviations. Further, it seemed best to formulate a few general rules to be followed in making abbreviations, rather than to compile a list of forms to be recommended.

The Committee decided to limit the subject more narrowly by considering only abbreviations to be used in the text, or general reading matter, and not those to be used in special matter, such as columns, box-headings, plates, figures, etc. The rules that follow are intended to apply to the text, and not primarily to such special matter. This Committee is of the opinion that it is impracticable to make general rules applicable to special matter; it believes that the rules herein stated should be followed as far as possible even in special matter, realizing, however, that clear-

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

ness is of the first importance, and that all rules must be secondary to that consideration.

Referring, then, to abbreviations in the text or general reading matter, the Committee recommends the observance of the following rules:

1. Use abbreviations only after nouns denoting a definite quantity. Example: "The power plant has a capacity of 10 h.p.," not "10 horse power;" but, "The capacity of the plant, in horse power, is ten."

2. Do not abbreviate abstract or descriptive words. Example: "horizontal return tubular boilers," not "h.r.t. boilers."

3. Use lower case characters for abbreviations. An exception to this rule may be made in the case of words spelled normally with a capital. Example: "B.t.u." and not "b.t.u." or "B.T.U." (British thermal unit), "U.S. gal." (United States gallon), "B. & S. gauge" (Brown and Sharp gauge).

4. Use a period after each abbreviation. In a compound abbreviation, do not use a space after the period. Example: "i.h.p." and not "i. h. p." (indicated horse power).

5. Use a hyphen to connect abbreviations in cases where the words would take a hyphen if written out in full. When a hyphen is used, omit the period immediately preceding the hyphen. Example: "3 kw-hr." and not "3 kw.-hr." (3 kilowatt-hours).

6. Use all abbreviations in the singular. Example: "17 lb." and not "17 lbs." (17 pounds), "14 in.," not 14 ins." (14 inches).

7. Never use "p." for "per," but spell out the word. Example: "100 ft-lb. per ton" (100 foot-pounds per ton); "60 miles per hr." (60 miles per hour).

8. Use decimals, as far as possible, in place of vulgar fractions. Example: "1.25 ft.," not "$1\frac{1}{4}$ ft."

9. In general, spell out an adjective qualifying the name of a unit. Example: "Boiler h.p." (boiler horse power). The exceptions to this rule are—"i.h.p." (indicated horse power), "e.h.p." (electric horse power), "b.h.p." (brake horse power), "e.m.f." (electromotive force), "m.m.f." (magnetomotive force).

10. Use "Fig.," not "Figure." Example: "Fig.3," and not "Figure 3."

11. In all decimal numbers having no units a cipher should be placed before the decimal point. Example: "0.32 lb.," not ".32 lb."

12. In the notation of large numbers, use "en" spaces instead of commas. Example: "1 520 125," not, "1,520,125."

13. Use the word "by" instead of "x" in giving dimensions. Example: "8 by 12 in.," not "8 x 12 in."

14. Never use the characters (′) or (″) to indicate either feet and inches, or minutes and seconds as periods of time.

The following forms are given as illustrations of these rules, and are recommended to be used:

NAME.	ABBREVIATION.
Inches	in.
Feet	ft.
Yards	yd.
Miles	spell out.
Pounds	lb.
Grains	gr.
Tons	spell out.
Gallons	gal.
Metres	m.
Millimetres	mm.
Centimetres	cm.
Kilometres	km.
Kilogrammes	kg.
Grammes	g.
Milligrammes	mg.
Kilogramme-metres	kg-m.
Metre-kilogrammes	m-kg.
Seconds	sec.
Minutes	min.
Hours	hr.
Linear	lin.
Square	sq.
Cubic	cu.
Per	spell out.
Fahrenheit	fahr.
Centigrade	cent.
Percentage	% or per cent.
Volts	spell out.
Ohms	spell out.
Watts	spell out.
Kilowatts	kw.
Kilowatt-hours	kw-hr.
Watt-hours	watt-hr.
Amperes	spell out.
Brake horse power	b.h.p.
Electric horse power	e.h.p.
Indicated horse power	i.h.p.
British thermal units	B.t.u.

NAME.	ABBREVIATION.
Gramme-calories	g-cal.
Kilogramme-calories	kg-cal.
Magnetomotive force	m.m.f.
Electromotive force	e.m.f.
Revolutions per minute	rev. per min.
Circular mils	cir. mils.
Miles per hour per second	miles per hr. per sec.
Candle-power	c-p.
Watts per candle-power	watts per c-p.
Mean effective pressure	spell out.
High pressure cylinder	spell out.
Diameter	spell out.

The members of this Committee have agreed to recommend the authorities in charge of the publications of the four national engineering societies to follow the rules given herein in their respective publications, as far as possible.

(Signed) CHARLES WARREN HUNT,
D. S. JACOBUS,
JOSEPH STRUTHERS,
CARY T. HUTCHINSON.

NOTE:—This report is presented herewith in its preliminary form and is open to further consideration, discussion, and change.

No. 1055.*

PRELIMINARY REPORT OF THE COMMITTEE APPOINTED TO SUGGEST A STANDARD TONNAGE BASIS FOR REFRIGERATION.

To The American Society of Mechanical Engineers:

Your committee appointed in December, 1903, to suggest "A Standard Tonnage Basis for Refrigeration" begs leave to report as follows:

1. Your committee concluded that a procedure similar to that adopted in establishing a unit for the boiler horse-power, and recommended by a committee of this Society in Volume VI., page 302, of the *Transactions*, be followed in this instance.

2. Your committee has made no attempt to include in this report other machines than those which perform refrigeration by evaporating a refrigerating fluid to a vapor.

3. Working on this line, the committee has confined itself strictly to a thermal rating and to the establishment of a set of conditions which would seem to represent good average engineering practice.

4. The subject was considered under two heads:

A. The selection of units to measure the cooling effect or the refrigeration produced.

B. The selection of a standard set of conditions under which a refrigerating machine, no matter what its type, shall be run in determining what is herein designated as its commercial tonnage capacity.

5. The unit adopted to measure the cooling effect, or the refrigeration, is the heat required to melt one pound of ice, which is 144 British thermal units,† and by dividing the refrigeration,

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† The figure used for the latent heat of fusion of ice in most reference books on refrigeration is 142 British thermal units. This figure comes from the experiments of Regnault, who obtained the value of 79.4 calories per kilogram. Per-

measured in British thermal units, by 144, the ice melting capacity in pounds is obtained. The unit for a ton (2,000 lbs.) of ice melting capacity is therefore 288,000 British thermal units. Your committee feel that this tonnage unit cannot be improved upon, and herewith recommend its adoption.

6. The commercial tonnage capacity is the refrigerating effect, expressed in tons of ice melting capacity produced by a machine in 24 hours when running continuously under the standard set of conditions.

7. Your committee appreciate the fact that it is very difficult to establish a standard set of conditions under which a refrigerating machine should be operated in order to determine its commercial capacity. Considering the matter from the standpoint of cost of plant and of steam and water economy, the best set of conditions to adopt seems to be those which often exists in ice making, namely, that the temperature of the saturated vapor at the point of liquification in the condenser, be 90 degrees Fahrenheit, and the temperature of evaporation of the liquid in the refrigerator be zero degrees Fahrenheit.

8. The ice making capacity is not the ice melting capacity of a machine, but is less, being usually about one-half the latter, because in making a pound of ice more refrigeration than 144 British thermal units is required owing to cooling the water to 32 degrees Fahrenheit, and certain unavoidable losses incident to the process.

9. The commercial tonnage capacity of any refrigerating machine using liquifiable vapor is based upon the actual weight of the refrigerating fluid that is circulated between the condenser and the refrigerator, and that is actually evaporated in the refrigerator

10. Under the conditions specified in paragraphs 7 and 9, twenty-five pounds of anhydrous ammonia per hour must be evaporated in the refrigerator for one ton of commercial tonnage capacity.* For other refrigerating fluids we do not at present

son determined it to be 80.0 calories, and Hess, 80.3 calories. Later experiments by Bunsen gave the value of 80 025, which is very nearly the mean of all the preceding values, and which is probably the most accurate. When reduced to British thermal units this value becomes 144 British thermal units per pound.

*This agrees closely with the results obtained by experiment when the cooling effect of the ammonia evaporated was checked by the refrigeration produced as actually measured.

make any recommendations as to the weight of the fluid that must be circulated.

11. The actual refrigerating capacity (in tons) of a machine may be determined from the quantity and range of temperature of brine, water or other secondary refrigerating liquid circulated as a refrigerant. The actual refrigerating capacity under the standard set of conditions should correspond closely to the commercial tonnage capacity.

12. We recommend that a committee be appointed to report, suggesting rules for the best methods of measuring the quantity of refrigerating fluid evaporated in the refrigerator, together with a code of rules for conducting tests of refrigerating machines or plants.

Respectfully submitted,

D. S. JACOBUS, *Chairman*,
E. F. MILLER,
P. D. C. BALL,
A. P. TRAUTWEIN,
GARDNER T. VOORHEES,
} *Committee.*

No. 1056.*

APPENDIX TO REPORT OF COMMITTEE ON STANDARDIZATION OF ENGINES AND DYNAMOS.

(Papers 887, May, 1901, and 916, December, 1901.)

At the meeting of the Society held in New York, December, 1901, the undersigned, a committee duly authorized, presented a report on certain questions connected with the Standardization of Engines and Dynamos. Experience since the recommendations of this Committee, have induced the members of that former committee to the belief that an improvement could be made by adding the following appendix.

They therefore ask that consent may be given to making this addition as an appendix to the report, and that authority may be given to distribute the appendix *with* the original report when the latter is called for. The appendix is as follows:

APPENDIX.

SUGGESTION F—SHAFT REDUCTION BEYOND ARMATURE FIT.

To facilitate pressing the armature upon the engine shaft, the engine builder should reduce the diameter of the shaft beyond the Armature Fit an amount *not less than* six thousandths of an inch.

JAMES B. STANWOOD, *Chairman*,
W. M. MCFARLAND,
A. L. ROHRER,
FRANK BALL,
W. B. FORBES.

To accomplish the purpose of the above signers, they move and second the passage of the following resolution:

Resolved, That the recommendation of Messrs. J. B. Stanwood, W. M. McFarland, A. L. Rohrer, Frank Ball and W. B. Forbes be treated as an appendix to the original report, and printed for circulation with that report, and to members of the Society, as the report shall be requested for future distribution.

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

No. 1057.*

COMPUTATION OF THE VALUES OF WATER POWERS, AND THE DAMAGES CAUSED BY THE DIVERSION OF WATER USED FOR POWER.

BY CHAS. T. MAIN, BOSTON, MASS.
(Member of the Society.)

Definition of Value.

1. The following definition of market value was given to the witnesses who were to testify on values in a recent important lawsuit.

2. "'Market value' means the fair value of the property, as between one who wants to purchase and one who wants to sell any article, not what could be obtained for it under peculiar circumstances, when a greater than its fair price could be obtained; not its speculative value; not a value obtained from the necessities of another. Nor, on the other hand, is it to be limited to that price which the property would bring when forced off at auction, under the hammer. It is what it would bring at a fair public sale, when one party wanted to sell and the other to buy."

Definition of Damage.

3. The definition of the damage due to the diversion of water was stated as, "The difference in market value, before and after the diversion."

Method of Determining Value.

4. The value of an undeveloped water power depends:

First. Upon its location, the amount and uniformity of flow, head, conditions affecting the cost of construction and transmis-

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

sion, use of exhaust steam and need of water for other purposes than power.

Second. Upon what the power is to be used for, whether for electric lighting and railway work, through most of the hours in the day with a variable load, for some use requiring a fairly steady load for twenty-four hours a day, or for running a textile mill or similar plant with a fairly steady load for about ten hours a day.

Third. Upon the market which can be served, whether it is secure and steady or must be built up and is somewhat unreliable.

5. The value of a privilege should be determined by comparison with the cost of producing power in such quantities and with such regularity as is required for the particular purpose for which it is to be used in a fairly economical manner at any place or places equally convenient for the transaction of the business under consideration. Some times the location is fixed, but oftentimes there can be a choice of locations.

6. In estimating the value of an undeveloped privilege, the steps followed are as follows:

(1) Determine the flow including the effect of storage and pondage.

(2) Determine the net head.

(3) Determine the horse-power which can be economically developed and used each month in an average year.

(4) Determine the minimum flow and power and from this the size of supplementary steam plant required if the power is to be developed above the minimum flow.

(5) Determine the shortage of water power during such months as there is a deficiency.

(6) Estimate the probable cost of development of the water power.

(7) Estimate the probable cost of the supplementary plant, using steam, gas, oil, or anything which is best for the location under consideration.

(8) Estimate the yearly cost of running the water power and supplementary plants, including the fixed charges on both, to produce a combined power suitable for the purpose for which the power is to be used.

(9) Estimate the cost of a steam, or other kind of plant, necessary to produce the power required.

(10) Estimate the yearly cost of running this plant, including fixed charges, to produce the power required.

(11) Subtract the cost of producing the power by water power and the supplementary plant from the cost of producing it by steam power, or some other method, alone. The difference, if positive, gives the apparent yearly saving by the use of water power. The apparent saving should be modified if necessary for location or any other thing affecting the value.

(12) Capitalize this difference at a rate which seems proper, and the result is the value of the privilege.

7. There seems to be a great difference of opinion as to the proper rate of capitalization, but in the purchase of water power privileges the buyer of his own free will assumes certain risks, as damages caused by freshets, changes of business, etc., which he will not assume for nothing. He is also basing his comparisons of cost of power upon the present cost of producing power, which cost may be reduced in the future. For these reasons, the yearly saving should be capitalized at a rate not less than 10 per cent.

8. Where a whole property is taken and the owner is free to move into an equal or more favorable location, the method and rate of capitalization given above should be used.

If the privilege is developed the total value includes the value of the plant.

9. The value of a plant will be its cost, less depreciation, up to the point where the cost of water power equals that of steam or some other power. Beyond this point, when water power costs more than steam power, the value of the improvements, although new, would not be represented by the cost but would be something less than the cost. It is the sum which could be paid for it new which would bring the total cost of water power including fixed charges down to the cost of steam power, less depreciation.

Method of Determining Damages.

10. The damage has been defined as the difference in value of the entire property before and after diversion.

11. It is usually unnecessary to go through an elaborate estimate of the value of the whole property, before and after the diversion, for the reason that many of the items of value will remain constant. The decrease in value, if there be any, is due to the fact that the running expense is increased by the diversion, and if this increased cost of running be capitalized at the proper rate the capitalized sum will represent the amount which the property is decreased in value, or the damage.

12. In estimating the damage to an undeveloped or abandoned power, the value before and after diversion should be estimated as described under the previous heading. The difference represents the damage.

13. If a privilege is developed and used, a valuable business carried on and a plant established which cannot be easily moved, the definition of damage still holds good, but in such a case it is customary to capitalize the yearly loss at a smaller rate than 10 per cent., as this damage is done against the owner's wishes, and as he should receive a sufficient sum from which, in his business or in some other way, he can obtain a sufficient income to make good his yearly loss. The writer has, unless otherwise instructed, capitalized the yearly loss at 5 per cent.

14. A privilege which produces a variable power and has no supplementary power is not damaged any more than if it were so supplemented, and it should be treated in the same way as though it were supplemented.

15. The writer has generally used the following method of determining the damage to an established property, due to the diversion of some of the water.

(1) Determine the flow, including the effect of storage and pondage, before and after the diversion.

(2) Determine the net head.

(3) Determine the horse-power which can be economically developed and used before and after diversion.

(4) The difference between the power used before and after diversion is the power diverted which causes damage.

(5) Estimate the additional yearly cost of running caused by the taking away of this power, of coal, attendance and supplies.

(6) If any permanent power has been taken, that is, power which can be relied upon in the lowest flow of the stream, estimate the cost of a steam plant or portion of plant necessary to make good the amount taken in the dry month.

(7) Estimate the fixed charges on this cost of additional supplementary plant.

(8) Add the extra cost of running and additional fixed charges and the sum represents the extra yearly expense.

(9) This extra expense capitalized at a proper rate represents the damage.

16. If it is necessary for the mills to maintain a steam plant of sufficient size to run the whole mill under the conditions exist-

ing before the taking, it is clearly not necessary for the defendant to furnish or maintain any further addition to the plant, and the damages consist of the increased expense of running the plant, already installed, due to the diversion.

17. If the total power required to run the mill is so large that the steam plant must be run all of the time, then there is no extra expense for attendance or supplies due to the diversion.

18. If the total power required is such that wheel plant can run the whole work for a portion of the year alone, and for the remainder must be supplemented by steam power, the time during which the engine must run may be extended by reason of the diversion, and in such case there is an addition to the expense of running for labor and supplies for such extra time, which should be added to the extra cost of coal, and the total extra expense capitalized at a proper rate will represent the damage.

Water Shed and Run-off.

19. Too much stress cannot be placed upon the importance of determining the flow of the stream under consideration. If careful gaugings have been made extending over considerable time, they are the most reliable information which can be had. If no gaugings have been made, an examination of the water shed should be made to ascertain its character, all existing rainfall records in the vicinity should be collected, and an estimate made of the run-off. Assistance may be had by comparison of similar rivers, the run-off of which is known.

20. The amount of data on the flow of streams which is available is increasing each year, as careful records are being kept on many rivers by persons or corporations who are interested in these matters, and by the United States Geological Survey.

21. The amount and uniformity of the run-off are two items which enter very largely into the value. The uniformity of flow depends largely upon the storage capacity and location of reservoirs on the water shed. The areas and capacities of such reservoirs should be ascertained and the net amount which can be drawn from them.

22. In estimating the average flow-off the months should be averaged in order of their dryness instead of in calendar order. If the flow is averaged by calendar months a great many irregularities in the flow are smoothed out and some of the flow is aver-

aged in which could not be held and used. By averaging all the dry months, all the second dryest, and so on, some of this evening up is eliminated, but it cannot be altogether avoided. The average flow averaged by months in the order of their dryness will be less uniform and nearer the truth than when arranged by calendar months.

23. The average year is the one used in estimating the available power or power diverted, but the effect during the year when the flow is less than the average must not be lost sight of.

Flow Used During Working Hours.

24. The flow at any given privilege is usually given in cubic feet per second for twenty-four hours a day and seven days a week.

25. If the power is used twenty-four hours a day and there are no disturbing influences above to break up the uniformity of flow during the whole day, a small mill pond will answer. If, however, there are mills above using all the water in ten hours a day, a large pond would be necessary to store and use it all in twenty-four hours.

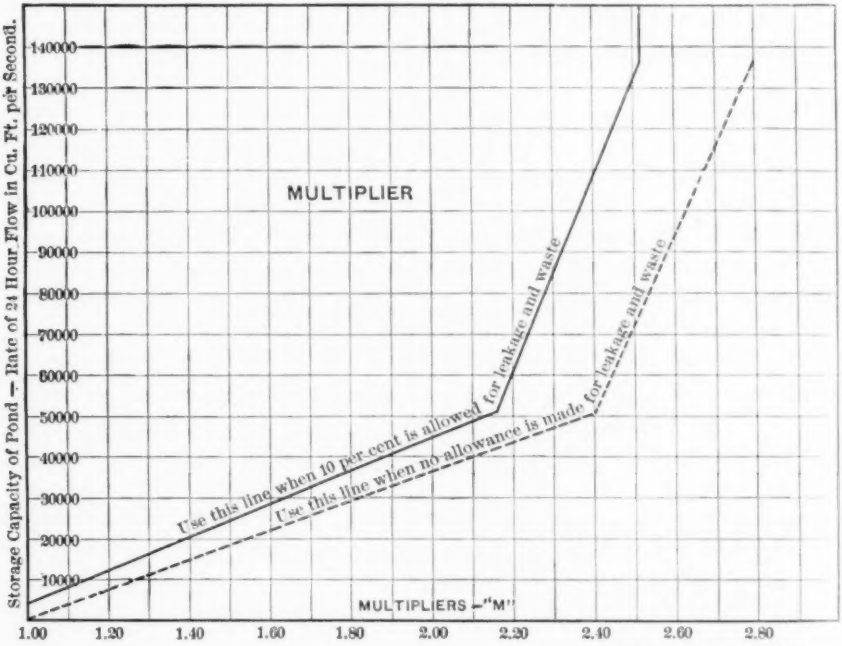
26. In a great majority of cases the water is used during the day for say ten hours a day and six days a week. If there is pondage enough so that it may be drawn down during the ten hours in the day enough to store the whole fourteen hour night flow, and if no water were wasted, the ratio of the flow used in ten hours to the 24-hour rate would be 2.4; that is, 2.4 times the 24-hour rate of flow would be used during the ten working hours of the day. If the pond could be drawn down Saturday, so that the night and Sunday flow could be stored, the ratio of 24-hour flow to that used in 10 hours a day and six days a week would be 2.8.

27. A certain amount of water is unavoidably wasted over the dam and by leakage through the various parts of the plant. This allowance of leakage and waste I usually place at 10 per cent. of the flow which could be theoretically stored and used. Using the above allowance for wastage, the maximum ratio of 24-hour flow to that which could be used in 10 hours, six days in a week, is $2.80 \times .90 = 2.52$.

28. When the pond cannot hold all of the flow during the time when the mill is not running the ratio will be something less than 2.52, and when no portion can be stored the ratio is 1. These ratios of amount of flow which can be used in 10 hours a

day to the total flow have sometimes been called "multipliers." They are the figures by which the 24-hour rate of flow is multiplied to get the rate which can be used during the working hours.

29. The multipliers are computed for an isolated privilege by adding to the cubic feet naturally flowing in ten hours the cubic



f=24 Hour Rate of Flow in C.F.S.
S=Storage in Cu.Ft.
M=$\frac{5}{12}$ the ratio, which the amount of Water which can be used 10 hours per day, 6 days in the week bears to the total amount flowing during the week.
Main, C.T.

When "M" is less than 2.16
"M"=$\frac{5}{10}\left(1+\frac{S}{36000f}\right)$

When "M" is greater than 2.1 but is less than 2.52.
"M"=$\frac{5}{12}\left(2.4+\frac{S-50400f}{21000f}\right)$

FIG. 1.

feet which can be stored each night, and to this adding one-sixth the cubic feet which can be stored during the 24 hours of Sunday, and dividing this sum by the cubic feet naturally flowing in ten hours.

30. With a series of mills, some with small ponds might be enjoying the benefits of larger ones above, although they might have no rights in them. If the location were directly below the one

with the larger pond so that the lower privilege practically takes the water as it comes from the upper one, it will get the benefit of all or nearly all of the storage above. As the distance increases between the two privileges, the length of time required for the water to get to the lower one would increase, and the benefit of the upper pondage would decrease. If it took one hour for the water to get down the multipliers would be 90 per cent. of those above, two hours 80 per cent., and so on, plus any storage of water from the water shed below the upper privilege which can be stored in the lower pond.

31. Where there is a series of mills the multipliers can be computed in this way until a privilege is reached, where the multipliers due to its own storage are greater than those obtained for anything above. This power then becomes the governing one for those below it until another is reached having large enough storage to establish a new set.

32. The computation of these multipliers is tedious, and in order to facilitate the computation I have worked out the formula for them, and have prepared a diagram which reduces the labor to a comparatively small amount:

Let f = 24-hour rate of flow in cu. ft. per sec.

s = storage in cubic feet.

M = $\frac{9}{10}$ of the ratio which the amount of water which can be used in ten hours a day, six days a week, bears to the total amount flowing during the week:

36,000 = number of seconds in 10 hours.

50,400 = " " " " 14 " or one night.

216,000 = " " " " 60 " " " working week.

2.16 = 90 per cent. of 2.4 ratio of 10-hour flow to 24-hour flow.

2.52 = 90 per cent. of 2.8 ratio of 10-hour flow to 24-hour flow + $\frac{1}{6}$ of Sunday or 4-hour flow.

When M is less than 2.16,

$$M = \tfrac{9}{10}\left(1 + \frac{s}{36{,}000\,f}\right).$$

When M is greater than 2.16 and less than 2.52,

$$M = \tfrac{9}{10}\left(2.4 + \frac{s - 50{,}400\,f}{216{,}000\,f}\right).$$

Fig. 1 shows the multipliers for various ratios of pondage to rate of 24-hour flow.

Use of the Multipliers.

33. The use of the multipliers is apparent when the problem is the determination of the amount of power which can be produced at a given place.

In estimating the damages caused by the diversion of a portion of the water shed, the power which can be produced before

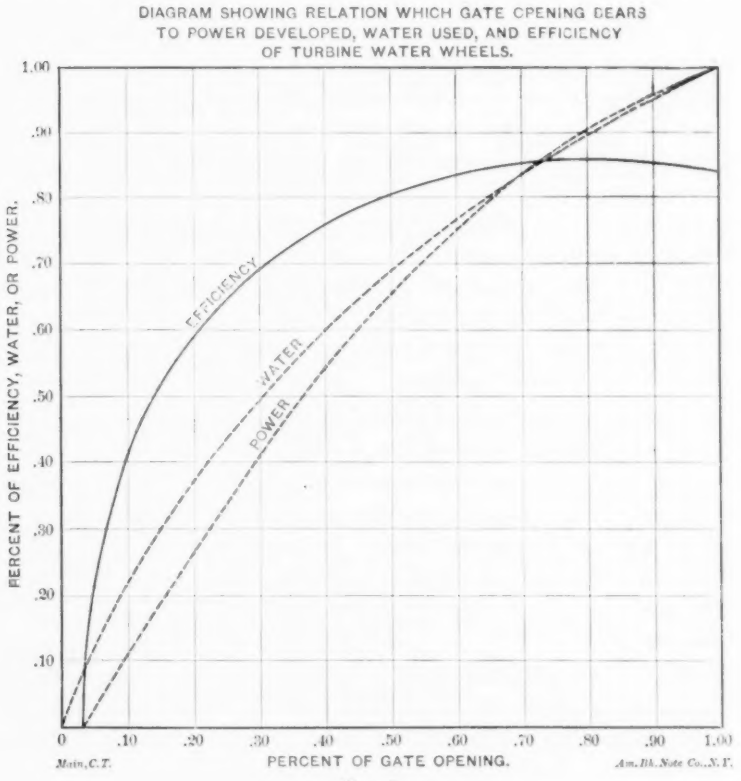


FIG. 2.

the diversion and the power which can be produced after diversion should also be estimated in the same manner, the difference between the two representing the amount of power diverted.

Head.

34. There may be several kinds of heads on the same development. There is the legal head, or the head to which the owner

has a right to develop his power. This may or may not have been developed to its full extent. It may be that the expense involved would be too great to warrant further development. In some cases it might be economy to make the expenditure necessary to get the benefit of some unused portion of the head.

35. The gross head is the head actually used for producing power and getting the water onto and away from the wheel.

36. The net effective head is the gross head minus the loss in head required to get the water onto and away from the wheel. This loss will vary with the length of the water ways leading to and away from the wheels, the velocity of the flowing water and the construction of such waterways.

37. In several manufacturing cities where the water power is controlled by a company which is separate from the mill owners, there is an allowance of one foot made from the gross head before charging for the water as used on the wheels.

38. The head should be measured with the wheels running. The only portion of the head which produces power is the difference in level directly above and below the wheel, when the wheel is running.

Efficiency of Wheels.

39. Some tests of water wheels show a maximum efficiency of about 85 per cent. It is probable that rarely over 80 per cent. is realized in practice after wheels have been installed for a short time, and this is for three-quarters to full-gate opening. When the gate opening is less than about three-quarters, the efficiency begins to drop.

40. Fig. 2 shows an efficiency curve which has been published by one of the large wheel makers in their catalogue as the result of tests on one of their wheels. This is an excellent curve and represents a wheel of maximum efficiency which is not often found in practice.

41. After wheels have been run for some time the buckets and guides are not as smooth as when they are new and the efficiency drops off. For these reasons I usually allow an average efficiency for wheels running under ordinary conditions of age, repair and variable gate opening of about 75 per cent. Under exceptionally good conditions and where there are several wheels this could be increased.

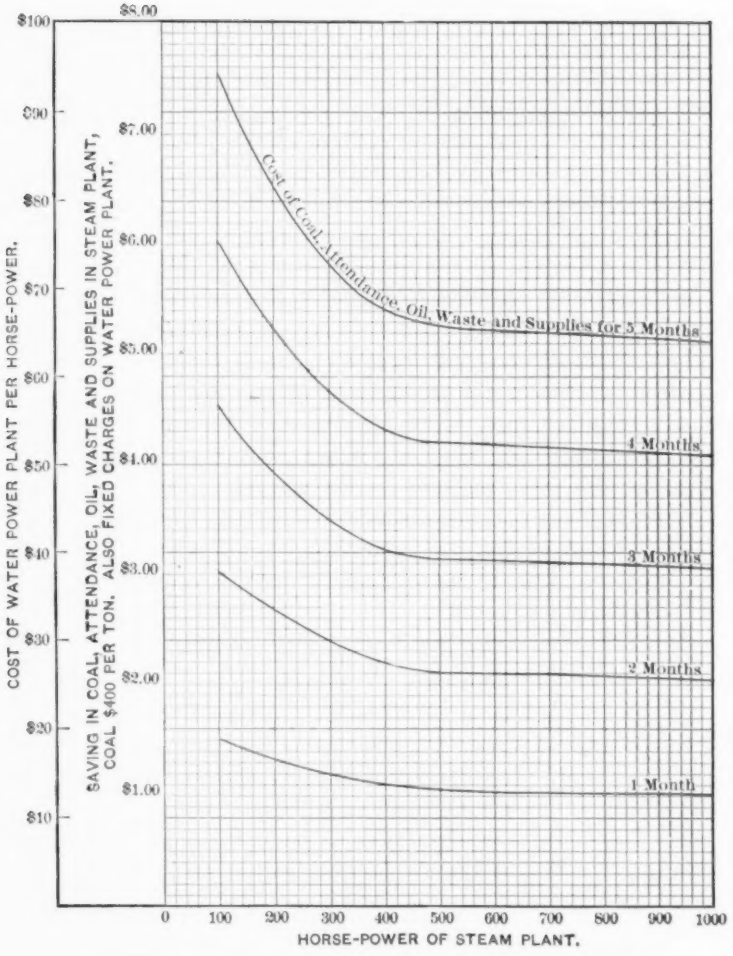


FIG. 3.

Limit of Low Flow.

42. With vertical wheels and bevel gears, and belt drive to head lengths of shafting, the friction losses are probably from 5 to 10 per cent. of the total. With horizontal wheels the friction losses are probably from 2 to 5 per cent.

43. On Fig. 2 there is plotted in addition to the efficiency the

percentage of water and power produced for different gate openings. From this it will be seen that with a small flow the efficiency and the amount of power developed will be small, and unless the total drainage area is fairly large, or the low flow is sustained from storage, the power developed in dry months may not be sufficient to run the wheel and overcome the frictional losses.

44. The flow required to produce 5 per cent. of the power is about 15 per cent. of the total water required to run the wheel full, and for 10 per cent. of power, 20 per cent of water is required. With one wheel only there must, therefore, be a flow of say 10 to 20 per cent. of the total used by the wheel to produce any useful power. With several wheels properly arranged this could be reduced somewhat. At such times of low flow the water must either be stored and used for a short time in the day or it will produce no useful work.

Effect of Back Water.

45. In a great many places there are periods during the year when the flow in the stream is so large that the water backs up below the wheel to a greater extent than the level of the water above the wheel can be raised, thus reducing the effective head and power. Sometimes the effect is so great as to prevent the use of the wheels.

Effect of Low Flow and Back Water.

46. The effect of low flow is to require an auxiliary power plant to make up the deficiency of water power, if it is necessary to run constantly, and if the flow drops so low as to produce no effective power, the auxiliary power plant must be of a capacity sufficient to run all of the work.

47. The effect of back water is to reduce the power produced by the water and to make it necessary to maintain a water power plant which has a surplus capacity in times of ordinary flow, or to maintain an auxiliary power plant and to run the same to make good the diminution of power if it is necessary to run full all the time. If the back water lasts for a long time and is so serious as to prevent any production of power from the wheels, the auxiliary plant must then be large enough to run the whole work.

Length of Time During Which Diversion Causes Damage.

48. In order to ascertain the difference in running expense due to diversion, it is necessary to know the average amount of power diverted and for how many months the diversion occurs. This latter can usually be ascertained by knowing the capacity

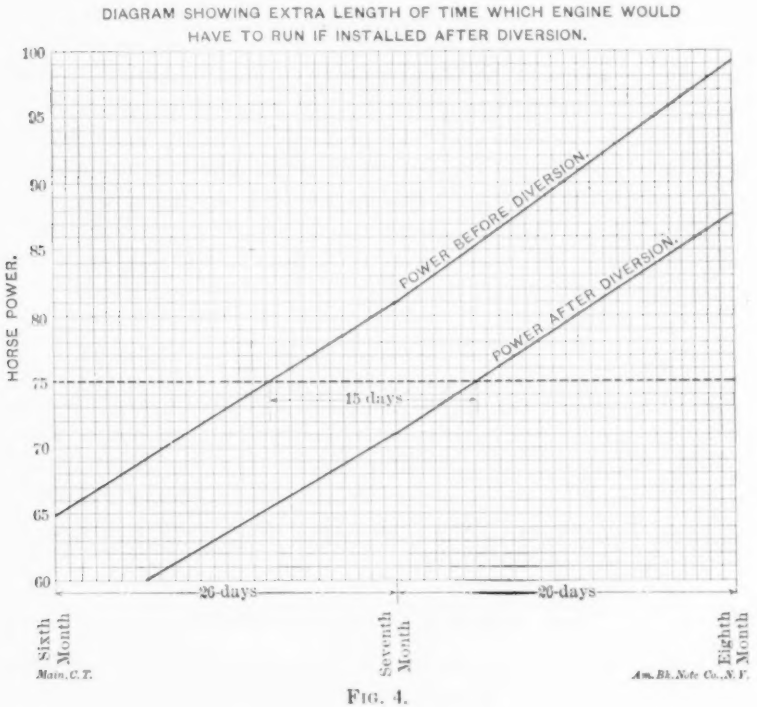


FIG. 4.

of the wheels in use, for in the majority of cases the wheel development will be an economical one.

49. It is sometimes the case, however, that larger wheels are installed than economy would warrant by overestimating the flow, or some other cause. Wheels are sometimes installed to be used in times of back water, remaining idle at other times. Where the variation of the use of power during the day is large, as for electric light and railway purposes, wheels may be installed for the peak load where pondage will allow this method of running. If the length of time when the diversion causes damage is measured by the capacity of the existing wheels, it

may appear to be for the entire twelve months of an average year. This cannot be true, for it would not pay to put in wheels to use all of the water in every month in the year. The diversion should be estimated for as many months as it would be economy to develop the power to use all the water under average conditions.

50. If the chance size of a wheel should be taken as measuring the length of time over which the damage continues, in a series of adjacent mills, some which had put in a portion of the wheel plant which could be used with economy would receive small damage, while a mill with a much larger wheel than would ordinarily be used would receive larger damages thereby, when in point of fact the damage would be the same, other things being equal, and with a series of mills the damages would not be proportioned properly unless the wheels were installed in each on the same basis of economical development.

Economical Development of Water Power.

51. In a large number of water power developments which I have examined, a very large percentage have been developed with wheel capacity sufficient to use all of the water from six to seven months in an average year, and during the remaining months water would go to waste. The economical development has been stated by some engineers to be nine months. No general statement is applicable to all conditions. It is a question of economics which requires solving for each particular case.

52. The factors which enter into the problem are on one side the cost of the water power development and the fixed charges on the same, plus the cost of water if anything is paid for it, and on the other side the saving which can be affected by the use of such a plant.

53. The cost of the dam will be a constant for any size of wheel development, other things being equal. The head gates, canal, racks, feeders, wheels, wheel-pits and tailraces must be increased in size and cost for the purpose of using a larger amount of water than the flow in the average month or sixth month of an average year, and the fixed charges for such increase in cost, plus the cost of water, represent the annual cost of the corresponding increase in water power.

54. The saving due to such increase in water power is repre-

sented by the saving in coal only on supplementary steam plant, necessarily run with such a varying water power, plus the cost of attendance and supplies on steam plant if it can be shut down entirely during the months of maximum power on the wheels. As the water power is increased in size to use water for a greater number of months, the cost of such increase for each additional month makes a saving for a less number of months, and there comes a time when the saving on steam power is less than the fixed charges on the additional cost of water power plant. Where these two items balance depends upon the following conditions:

(1) Cost of running the water power plant for each increment of power.

(2) Saving affected by the decreased use of steam power.

55. The variation in the cost of the water power plant per horse-power is very large. The principal causes for this are the variation in head and distance from the source of supply of the water to the point of discharge. The cost of construction will also vary with local conditions.

56. The saving affected would also vary largely, depending principally upon the number of hours run during the day, the cost of coal, and whether by increasing the size of water power plant the auxiliary power plant could be stopped during the months in which the water power was producing full load.

57. An example will suffice to make this clear. Supposing the cost for each additional horse-power of water power plant required to use all the water for a longer period was $60 a horse-power. The fixed charges on this, including interest, will be not less than 8 per cent., or $4.80 per year. The cost of coal and attendance on a steam plant of say 500 horse-power, when running ten hours a day, with coal at $4 per ton, is about $13 per year per horse-power, or $1.08 per month. $4.80 ÷ $1.08 = 4.44 months. In other words, it would not pay to develop such a power to use all the water for more than about seven and one-half months.

58. If the engine or boilers cannot be shut down at all, a less saving could be made and the power could be economically developed for a less period than seven months.

59. The various conditions and lengths of time required to have the saving equal the fixed charges are shown on Fig. 3. This diagram is figured on coal at $4 per ton, and with a running time

TABLE ..... NAME ....

Col. 1	Flow and Power before Diversion.								Flow and Power after Diversion.						Power Diverted.
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	
Months in the order of their dryness.	Natural flow in cu. ft. per sec. on one square mile.	Total flow cu. ft. per sec. = Col. 2 × square miles.	Multiplier or ratio of flow which can be used in 10 hours to 24-hour rate.	Total flow which can be used in 10 hours a day = (Col. 3 × Col. 4.	H.-P. which can be used in 10 hours a day on one foot fall = Col. 5 × .0851.	Total H.-P. which can be used in 10 hours a day = (Col. 6 × ft. head.	H.-P. which present wheels can use or which would or-dinarily be developed.	Total flow cu. ft. per sec. = Col. 2 × square miles.	Multiplier or ratio of flow which can be used in 10 hours to 24 hour rate.	Total flow which can be used in 10 hours a day = Col. 9 × Col. 10.	H.-P. which can be used in 10 hours on one foot fall = Col. 11 × .0851.	Total H.-P. which can be used 10 hours a day = Col. 12 × ft. head.	H.-P. which present wheels can use or which would or-dinarily be developed.	H.-P. diverted which causes damage = (Col. 8 − Col. 14.	
Driest.....															
2d "															
3d "															
4th "															
5th "															
6th "															
7th "															
8th "															
9th "															
10th "															
11th "															
Wettest.....															
Totals and Averages...															

Average H.-P. diverted ..... for ..... months in an average year .....

of ten hours a day, six days a week. Similar diagrams could be made for any other prices of coal and time of running.

60. To use the diagram, supposing the water power plant cost $50 per horse-power, and the size of steam plant is 200 horse-power. On the ordinates find $50 cost of water power plant. Run along horizontally until this line intersects the vertical line of 200 horse-power of steam plant, and these two lines will be found to intersect about on the curve marked three months.

If the water plant cost $70 and the steam plant were 350 horse-power, the time is five months during which water should waste.

Table Showing Flow and Power.

61. The table on page 83 shows a convenient form for tabulating the flow and power. The first half is useful in estimating the value of a privilege, and the whole table for estimating damages when a portion of the flow is diverted.

62. The only thing needing explanation is the figure .0851 which appears in the headings of columns 6 and 12. This is the horse-power produced by one cubic foot of water per second on one foot head with an efficiency of 75 per cent. With 80 per cent. efficiency the figure is about .091.

Approximate Cost of Water Power Development.

63. In connection with the preceding diagram is printed a table, page 85, showing approximately the cost per horse-power of water power plants, not including dam, canal and buildings, for different heads, distances from feeder head to end of tailrace, and horse-power of development.

It is not expected that this table will cover all cases, but it will give approximate figures for ordinary conditions, and is useful in making rough preliminary estimates.

Extra Length of Time Which Steam Plant Must Run on Account of Diversion.

64. If the power of the stream is worked out before and after diversion, it will probably be found that the auxiliary power plant must be run for a longer time after the diversion.

65. This would be shown in full months in the tables showing

TABLE OF ESTIMATED COSTS PER HORSE-POWER OF WATER POWER PLANTS

Having Horizontal Turbines, Steel Penstocks, and Walled Tailraces —Dam and Buildings not Included.

	"L".	10-ft. fall.	15-ft. fall.	20-ft. fall.	30 ft. fall.	40-ft. fall.
1,000 H.-P.	100 feet	$65.14	$40.92	$29.37	$19.40	$14.59
	200 "	71.91	45.48	32.84	21.70	16.38
	300 "	78.66	50.05	36.30	24.01	18.17
	400 "	85.43	54.62	39.77	26.32	19.95
	500 "	92.20	59.18	43.23	28.63	21.74
	600 "	98.96	63.75	46.69	30.94	23.53
900 H.-P.	100 "	65.35	41.00	29.56	19.56	14.79
	200 "	72.08	45.58	33.04	21.85	16.60
	300 "	78.80	50.16	36.53	24.14	18.41
	400 "	85.50	54.74	40.01	26.41	20.22
	500 "	92.24	59.32	43.49	28.70	22.02
	600 "	98.96	63.91	46.97	30.99	23.84
800 H.-P.	100 "	65.48	41.10	29.65	19.68	14.99
	200 "	72.22	45.68	33.12	21.98	16.78
	300 "	78.95	50.26	36.68	24.28	18.57
	400 "	85.70	54.84	40.04	26.56	20.35
	500 "	92 43	59 41	43.52	28 87	22.14
	600 "	99.15	64.00	46.98	31.17	23.93
700 H. P.	100 "	65.72	41.19	29.87	19.90	15.12
	200 "	72.48	45.75	33.35	22.19	16.93
	300 "	79.23	50 30	36.82	24.49	18.73
	400 "	86.00	54.86	40.29	26.77	20.54
	500 "	92.74	59.42	43 77	29.07	22 34
	600 "	99.50	63.97	47 25	31.37	24.16
600 H. P.	100 "	65.86	41.56	30.00	20 02	15.37
	200 "	72.64	46.14	33.49	22.34	17.20
	300 "	79.42	50.72	36.97	24.64	19.04
	400 "	86.20	55 30	40.45	26.95	20.87
	500 "	92.98	59.88	43.94	29.27	22.70
	600 "	99.76	64 47	47.42	31 57	24.54
500 H.-P.	100 "	66.00	41 70	30.24	20.24	15.52
	200 "	72.82	46.32	33.78	22.56	17.31
	300 "	79 64	50.94	37.28	24.86	19 10
	400 "	86.46	55.56	40.80	27.16	20.88
	500 "	93 28	60.16	44.34	29.48	22.66
	600 "	100.10	64.80	47.84	31.80	24.44
400 H.-P.	100 "	66.28	42.03	30.55	20.79	16.00
	200 "	73.16	46.65	34.05	23 10	17.82
	300 "	80.03	51.26	37.53	25.40	19.64
	400 "	86.90	55 88	41.03	27 73	21.45
	500 "	93 78	60.50	44.53	30.03	23.27
	600 "	100.65	65.13	48.03	32.35	25.08
300 H.-P.	100 "	66 87	42.67	31.09	21.40	16.70
	200 "	73.70	47.30	34.57	23.83	18.33
	300 "	80.54	51.94	38.07	26.18	20 16
	400 "	87.33	56.54	41.54	28.53	22.00
	500 "	94.17	61 18	45.04	30.88	23.83
	600 "	101.00	65.78	48.51	33.22	25.67
200 H.-P.	100 "	68.50	44.22	32.45	22 61	17.60
	200 "	75.35	48.84	35.97	24.97	19.47
	300 "	82.25	53.45	40.04	27 33	21.34
	400 "	89.10	58.10	43.56	29.70	23.21
	500 "	96.00	62.70	47.08	32.06	25.08
	600 "	102.85	67.35	50.60	34.43	26.95
100 H.-P.	100 "	71.39	46.64	34.76	24.75	19.80
	200 "	78.43	51.37	38.39	27.17	22.00
	300 "	85.47	56.10	42.02	29.59	24.20
	400 "	92.51	60.83	45.65	32.01	26.40
	500 "	99.55	65.56	49.28	34.43	28.60
	600 "	106.60	70.28	52.91	36.85	30 80

Note: "L"—Distance from Feeder Head to end of Tailrace. Cost of Canal, if any, not included.

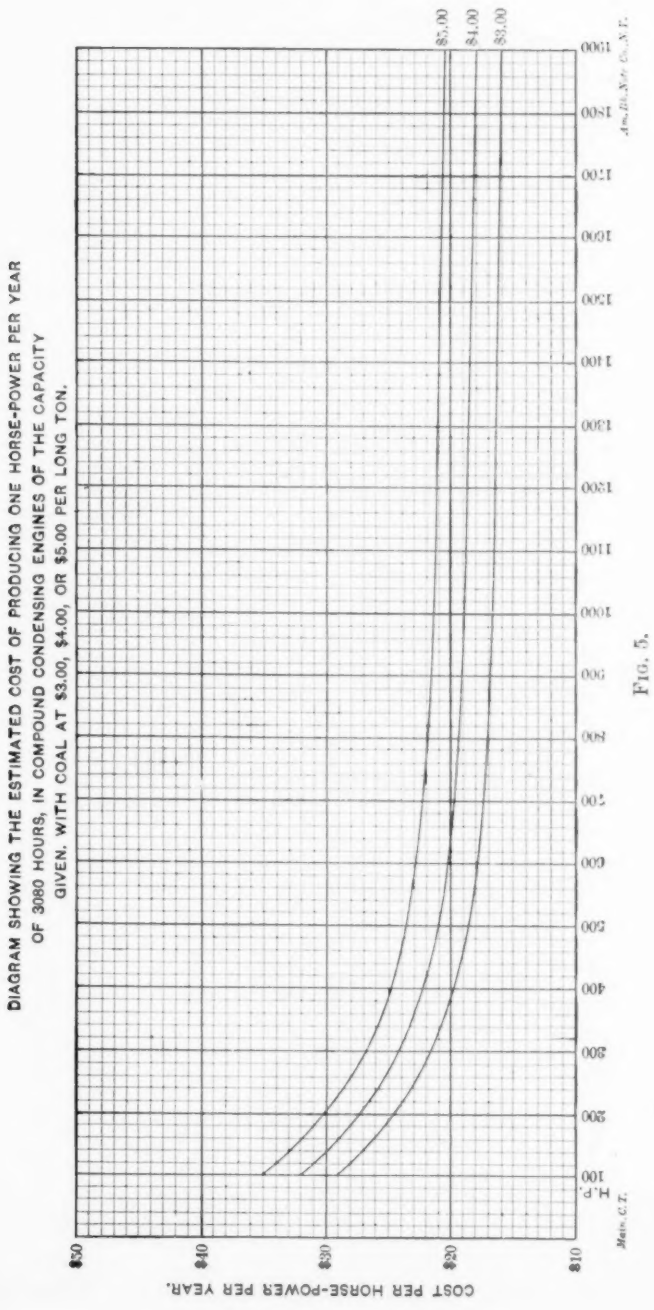


DIAGRAM SHOWING THE ESTIMATED COST OF PRODUCING ONE HORSE-POWER PER YEAR
OF 3080 HOURS, IN COMPOUND CONDENSING ENGINES OF THE CAPACITY
GIVEN. WITH COAL AT $3.00, $4.00, OR $5.00 PER LONG TON.
$5.00
$4.00
$3.00
H.P.
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
1700
1800
1900
$50
$40
$30
$20
$10
COST PER HORSE-POWER PER YEAR.

Fig. 5.

the power before and after diversion, but it would not show fractions of a month.

66. The method of ascertaining the extra length of time is shown in Fig. 4. The ordinates show horse-power which the whole stream can produce and the abscissæ the number of working days. By plotting the horse-power which can be developed before and after diversion and drawing diagonal lines from month to month, the extra number of days when the auxiliary plant must run is shown where the diagonal lines cross the horizontal line of wheel development.

Steam or Other Power Plant to be Used in Making Good Power Diverted.

67. In order to determine the damage it is necessary to estimate the cost of replacing the power taken away, not necessarily by the auxiliary plant already existing at the mill, for such a plant may be an extremely uneconomical one, and the mill which had the poorest plant would get the most damages, but by a *fairly economical plant of the size and character such as the business under consideration would naturally use.* Unless this method is pursued with a series of mills, the one which had put itself into the best shape would receive a smaller amount of damage than its neighbor where things were in bad shape.

68. It is improbable that a concern would go on putting in each time it renews its plant an uneconomical plant, and it is highly probable that the low grade engine will be improved in efficiency.

As the damages are figured forever, it will add very little to the damages for the comparatively short time which the uneconomical plant will be obliged to run.

Privileges with No Auxiliary Plant.

69. The basis on which the damages should be estimated in these cases where there is no auxiliary power plant is the same as for any other. The mere chance that an owner can manage in some way to run his business in accordance with the fluctuating flow of the stream, does not entitle him to any greater damage than his neighbor, who is fitted up to run continuously. A small amount of power diverted will not make the conditions enough worse in most cases to require the addition of a supplementary plant for that reason alone. An allowance, however, should be

made in estimating the damages in such cases for such portion of a steam plant as would ordinarily be installed to produce a

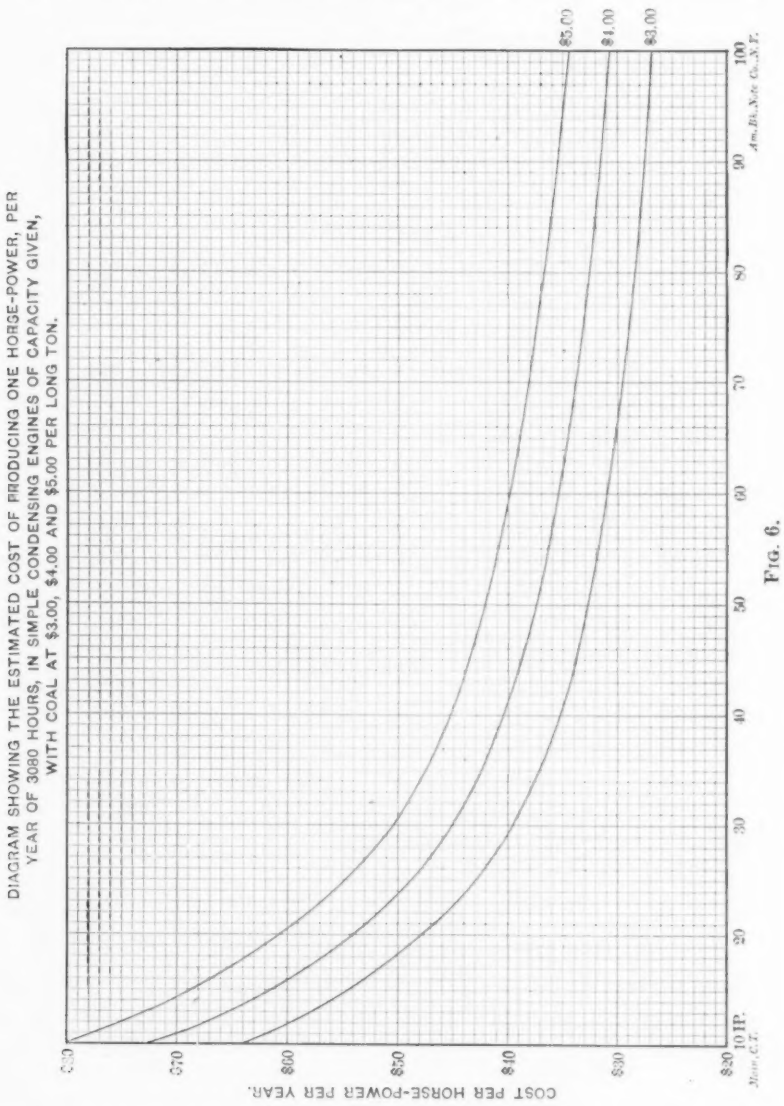


FIG. 6.

uniform power as is equal to the power diverted in the dry month, and the running expenses should be estimated on a plant of full size and not on a very small plant.

70. The proper method of ascertaining the damages to a plant of this sort is to find the difference in value before and after diversion, and this is obtained by finding the cost of producing a uniform power before and after diversion.

Allowance for Permanent Power Diverted.

71. If any power is diverted which can be depended upon all the time, an allowance should be made for this. If the diversion is comparatively small the fixed charges should be allowed on the cost of a portion of the large plant equivalent to the amount of power diverted. Thus, supposing 5 horse-power is diverted in the dry month, and the cost of the steam plant is $60 per horse-power. $5 \times \$60 = \$300$. $\$300$ @ $12\% = \$36$. This capitalized at $5\% = \$720$.

If this allowance is made the owner may increase the capacity of his plant by 5 horse-power when he renews it, and will have been recompensed for this expense.

If the interest charges are not included in the fixed charges, the cost of plant, $300, should be added to the capitalized sum.

If the diversion is a comparatively large amount, it may be necessary to remodel and increase the existing steam plant, or to put in a new one. Allowance should be made in the same way for this.

Cost of Steam Power.

72. The cost of steam power usually has an important bearing upon the settlement of damages. The accompanying Figs. 5, 6 and 7 have been prepared, which show the yearly cost of producing steam power under various conditions and cost of coal when running ten hours a day and six days a week with a fairly steady load. They are intended to show the expense of running under everyday conditions on such a plant as a prudent man would install and run with ordinary skill.

The cost of 24-hour power for 365 days a year is about 2.2 times the cost for 10-hour power for 308 days.

The cost of 24-hour variable load cannot be stated without knowing all the conditions.

Coal Consumption Used In Estimating Damages.

73. The coal consumption used in estimating damages when the power diverted must be made good under a varying load contingent

upon the fluctuation of the water power, should be somewhat larger than the coal consumption for a fairly steady load. I have usually added about 20 per cent. to the coal consumption required for a

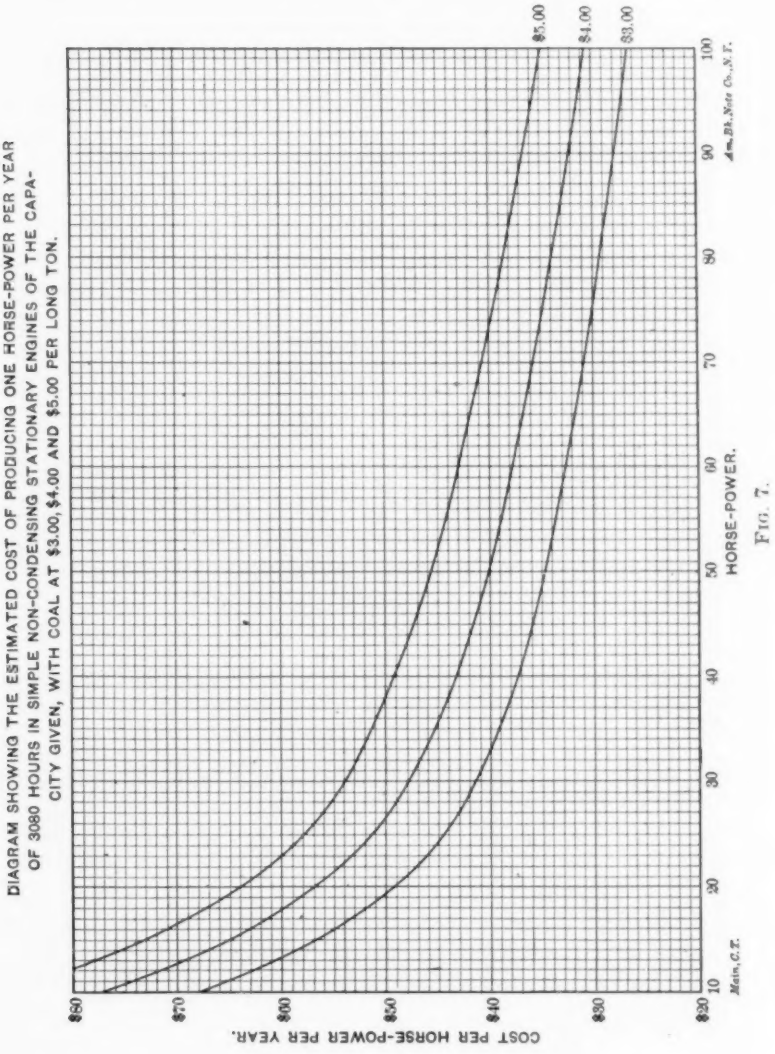


FIG. 7.

steady load. The fluctuation of the water power will usually not be great for a single day, but the variation covers longer periods, as weeks or months.

Effect of Use of Steam for Other Purposes.

74. In many textile and other mills low-pressure steam and hot water can be used in the manufacturing processes, and for warming the buildings. The amount varies largely, in some cases being more than the equivalent amount of steam exhausted from an engine large enough to run the work, or to the amount of water required for condensing for an engine of the same size. Rarely ever in a textile mill would the amount fall below 20 per cent. of the total heat rejected by the engine. This has the effect of reducing the value of water power for such mills, and has the effect of reducing the damages as figured on straight power conditions.

75. It is never attempted, however, to estimate this effect in suits for damages. It should be considered in estimating values, and it has its effect upon the selling value of water powers, making them of less value for industries having use for low-pressure steam and warm water.

76. All of the varying conditions of different industries have an effect of producing a sort of average selling value for water powers, but each case requires examination and estimates of its own.

No. 1058.*

THE FORCING CAPACITY OF FIRE TUBE BOILERS.

BY F. W. DEAN, BOSTON, MASS.

(Member of the Society.)

1. There is a widespread impression among many persons interested in boilers that the fire tube type of boiler has less capacity to be worked above its rated power than boilers of the water tube type and that it is necessary to provide such boilers with more heating surface per horse-power than water tube boilers require. It is pointed out that for many years it has been customary to base the horse-power of horizontal return tubular boilers upon 15 square feet of heating surface per horse-power, although at present some makers of this boiler venture to base it upon 12 square feet.

2. It is important to remember that the rating of 15 square feet was in use long before this Society established the meaning and measure of a boiler horse-power, and by persons who probably knew very little about boiler performance and the capacity of heating surface to absorb heat. It probably had been found at some time, that, with the prevailing type of engine in use it was always safe to provide boiler power upon this basis. It had not been customary to make boiler tests at that time as frequently as now, and no exact knowledge of boiler performance existed, or, if it did, it was confined to comparatively few persons.

3. The introduction of water tube boilers had caused more intelligent study of boiler phenomena, and there are now associated with the builders of this type of boiler many of the brightest men and most energetic salesmen.

4. It is the object of this paper to show that the prevailing opinion upon the relative forcing capabilities of the two types of boiler is incorrect, and in support of this claim I shall give par-

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ticulars of several boiler tests upon different kinds of fire tube boilers, some made by others and some by myself.

5. Before proceeding with the details, however, I will briefly mention the reasons given for the alleged superiority of the water tube boiler in the respect under consideration. It may also be well to state what is meant by ability to stand forcing. In a general way this ability is the capacity to evaporate much more water than the rated power of the boiler would require. This amount is often 20, 50, or even 100 per cent. in excess of the rated requirements, and depends upon the amount of fuel consumed in a unit of time. This in turn depends chiefly upon the draft in the fire box and the freedom with which air can pass through the grate or stoker and through the fuel itself.

6. An alleged reason for the superiority is that the water tube boiler contains less water than the fire tube boiler, by which is meant, I suppose, that it contains less water in proportion to the heating surface. I have been able to ascertain the amount of water contained by a few water tube boilers, and have computed that contained by several fire tube boilers, drawings of which I have. The results are as follows:

Horizontal Water Tube boiler				A.....	7.66	lb.	per	sq.	ft. of heating	surface.
"	"	"	"	B.....	8.31	"	"	"	"	"
"	"	"	"	C.....	8.62	"	"	"	"	"
"	"	"	"	D.....	8.37	"	"	"	"	"
Vertical	"	"	"	.....	10.30	"	"	"	"	"
Semi-vertical	"	"	"	.....	11.42	"	"	"	"	"
90 in. hor.	"	"	"	.....	7.74	"	"	"	"	"
84 " "	"	"	"	.....	7.24	"	"	"	"	"
78 " "	"	"	"	.....	7.00	"	"	"	"	"
72 " "	"	"	"	.....	6.73	"	"	"	"	"
Manning vertical		"	"	.....	7.14	"	"	"	total	"
Corliss	"	"	"	.....	11.00	"	"	"	"	"

Some vertical boilers of my own design contain more water than the Corliss.

7. An inspection of this table as far as it goes shows that water tube boilers generally contain more water than horizontal fire tube boilers. It appears from this that whatever advantage accrues from small water contents lies with the horizontal return tubular boiler. The vertical water tube boiler, as well as some designs of fire tube boilers, contains more water than horizontal boilers.

8. The comparison of horizontal boilers reveals conditions that

should not be surprising, for water tube boilers are made up of drums, headers and tubes, all of which contain considerable water, while the horizontal return tubular boiler has most of its water space filled with tubes, especially when there is no manhole in the front head below the tubes.

9. A little consideration, however, will show that the quantity of water contained by a boiler has no effect upon forcing capacity after steam is once up to the working pressure. The reason for this is that heat added cannot raise the temperature of the water, and can, therefore, only increase the quantity of steam formed and increase the quantity and temperature of the escaping gases. It therefore appears that both types of boiler have equal merits in this respect.

10. Another claim is that as the flame enters among the tubes of water tube boilers the evaporation is forced more than if it did not reach them. As flame is combustion in process, and is likely to be prematurely terminated by contact with the relatively cold surfaces of the tubes, it is apparent that it is disadvantageous to have the flame pass among the tubes.

11. Another claim is that the hot gases impinge against water tubes and therefore give up more heat than they would if they passed parallel to them. The fallacy of this argument is apparent when it is remembered that gases give up heat because their temperature is higher than that of the heat receiving medium, and not because they are thrown against this medium. Moreover, some kinds of water tube boiler compel the gases to move parallel to the tubes, and this feature is held to be highly advantageous by interested persons. Incidentally it will be found disadvantageous to project boiler gases against the outsides of tubes, for examination shows that it causes wedge shaped accumulations of soot at the very part of the tubes where the greatest reception of heat was expected, thus rendering this part the least effective of any.

12. I believe it is also claimed that there is less reduction of draft between the smoke flue and the fire box in water tube than in fire tube boilers on account of larger spaces between the tubes of the former than through the tubes of the latter. Whether this is true or not I cannot say in general, but in cases where I have determined it the results are inconsistent and contradictory. I have found, however, that the vertical fire tube boiler requires the least draft for burning coal at a given rate. This, I think,

is due to the path of the air being at right angles to the fuel and therefore shorter and less impeded by the fuel than with horizontal boilers, in which the path of the air is diagonally through the fire. In this connection, observe the slight draft found in some cases farther on.

Horizontal Return Tubular Boilers at the Atlantic Mills, Lawrence, Mass.

Referring to the results of trials, there were in 1894, at the Atlantic Mills, Lawrence, Mass., several horizontal return tubular boilers 24 years old containing their original tubes, and which had never been cleaned on their water sides. The following are the general dimensions of each boiler:

Diameter of shell	60	in.
Diameter of tubes	3½	"
Length of tubes	20	ft.
Number of tubes	55	
Heating surface, fire sides	1,112	sq. ft.
Grate area, 1st test, 21.77 sq. ft. 2d test,	21.51	"

I tested one of these boilers on November 22, and again on December 18, 1894, with the following general results:

	1st Test.	2nd Test.
Duration	10 hours.	10.18 hours.
Kind of coal	Georges Creek Cumberland.	
Calorific value of coal by analysis	13,916 B. t. u.	14,360 B. t. u.
Steam pressure, average	84.03 lb.	84.7 lb.
Force of draft in smoke box	0.21 in.	0.25 in.
Temperature of feed water	67 degrees	55 degrees
" " escaping gases	373 degrees	389 degrees
Dry coal consumed including wood equivalent	4,576 lb.	4,746 lb.
Total water evaporated from and at 212 degrees	47,863 "	50,841 "
Water evaporated per pound of dry coal from and at 212 degrees	10.46 "	10.71
Water evaporated per pound of combustible from and at 212 degrees	11.18 "	11.57 "
Commercial horse-power	138.7	144.8
Horse-power per square foot of grate	6.37	6.73
Heating surface per horse-power	8.00 sq. ft.	7.70 sq. ft
Dry coal burned per square foot of grate surface per hour	21.4 lb.	21.7 lb.
Dry coal burned per square foot of heating surface per hour	0.41 "	0.42 "
Water evaporated per square foot of heating surface per hour, from and at 212 degrees	4.30 "	4.49 "

It will be observed from these results that a horse-power was obtained from 8 and 7.7 square feet of heating surface, that 6.37 and 6.73 horse-power were obtained from each square foot of grate, that 0.41 pounds and 0.42 pounds of coal were consumed per square foot of heating surface per hour, and that these results were obtained with not over $\frac{1}{4}$ inch of draft. These results are quite remarkable, especially as the economy was not very inferior, and especially considering the age of the boiler.

Horizontal Return Tubular Boilers at S. D. Warren & Company's Paper Mill.

At the paper mill of S. D. Warren & Company, Cumberland Mills, Maine, there are six 90-inch horizontal return tubular boilers, each containing 4,027 square feet of heating surface. When the first four were put in two of them were equipped with Babcock & Wilcox chain grates and two with second-hand shaking grates for temporary use.

The general dimensions are as follows:

Diameter of shell	90	in.
Diameter of tubes	3	"
Length of tubes	20	ft.
Number of tubes	258	
Heating surface, fire sides	4,027	sq. ft.
Grate area with chain grate stoker	61.87	"
" " " hand fired grate	48.75	"

I tested one of the hand fired boilers on April 4, and one with the stoker on April 5, 1900, with the following general results:

	Hand.	Stoker.
Duration	9.5 hours.	8.75 hours.
Kind of coal	Georges Creek Cumberland.	
Calorific value of coal by analysis	13,961 B. t. u.	
Average steam pressure	92.1 lb.	94 lb.
Force of draft in smoke box	0.90 in.	0.91 in.
Temperature of feed water	85.5 degrees	85.5 degrees
" " escaping gases	569 degrees	645 degrees
Dry coal consumed	19,522 lb.	20,351 lb.
Per cent. of refuse in ash pit	8.27%	8.72%
Total water evaporated from and at 212 degrees	206,437 lb.	223,983 lb.
Water evaporated per pound of dry coal from and at 212 degrees	10.75 "	11.06 "

	Hand.	Stoker.
Water evaporated per pound of combustible from and at 212 degrees	11.53 lb.	12.06 lb.
Commercial horse-power	630 h. p.	742 h. p.
Horse-power per square foot of grate	12.92 "	12.00 "
Heating surface per horse-power	6.39 sq. ft.	5.43 sq. ft.
Dry coal burned per square foot of grate per hour	42.2 lb.	37.6 lb.
Dry coal burned per square foot of heating surface per hour	0.51 "	0.58 "
Water evaporate per square foot of heating surface per hour from and at 212 degrees.	5.39 "	6.35 "

The rate of combustion, the temperature of the escaping gases, the small heating surface per horse-power, the amount of coal burned and the horse-power developed per square foot of grate, and the amount of coal burned and water evaporated per square foot of heating surface are truly remarkable and are seldom equalled. They show that this type of boiler is as well suited to forcing as any, and leave nothing to be desired. The water used was determined for each hour, and during one hour the horse-power of one boiler was 794, and with the other 793, or about five square feet of heating surface per horse-power. It should be noticed also that the temperatures of the escaping gases are remarkably low considering the quantity of coal burned per square foot of heating surface. The maximum flue temperature was on the hand fired test 630 degrees and on the stoker test 700 degrees.

Manning Vertical Boiler at the Bristol Manufacturing Company's.

On March 1, 1894, Professor J. E. Denton tested a Manning vertical fire tube boiler at the mill of the Bristol Manufacturing Company, New Bedford, Mass. The dimensions and general results follow:

Diameter of shell	60 in.
Diameter of fire box	6 ft.
Height of fire box	42 in.
Number of tubes	180
Diameter of tubes	2½ in.
Length of tubes	15 ft.
Grate surface	28.7 sq. ft.
Water heating surface	1,383 "
Superheating surface	471 "
Grates	fixed hand fired.

General Results of Test.

Duration	9.91 hours.
Kind of coal	Pocahontas.
Calorific value of coal by calorimeter	14,675 B. t. u.
Average steam pressure	122.2 lb.
Temperature of feed water	38.1 degrees
" " escaping gases	531.4 degrees
Dry coal consumed	6,460 lb.
Per cent. of refuse in ash pit	9.63%
Water evaporated per hour from and at 212 degrees	6,679 lb.
" " per pound of dry coal from and at 212 degrees	10.03 "
" " " " " combustible from and at 212 degrees	12.00 "
Commercial horse-power	194 h. p.
Horse-power per square foot of grate	6.62 "
Water heating surface per horse-power	7.13 sq. ft.
Dry coal burned square foot of grate per hour	23 lb.
" " " " " " water heating surface per hour	0.47 "
Water evaporated per square foot of water heating surface per hour from and at 212 degrees	4.83 "

This is not extreme forcing, but the results are quite surprising. For instance, the submerged length of the tubes was about 11 feet, and the evaporation was at the high rate of 4.83 pounds of water from and at 212 degrees, per square foot of water heating surface. The evaporation per pound of combustible was at the unusual rate of 12 pounds from and at 212 degrees. This shows that vertical tubes are efficient absorbers of heat.

Manning Boilers at the Burgess Sulphite Fibre Company's.

Two remarkable tests of Manning boilers which were equipped with Roney stokers were made at the mill of the Burgess Sulphite Fibre Co., Berlin, N. H., and of these one in which a draft of 1.28 inches of water was used is unique. The general particulars are given below:

Diameter of tubes	2½ in.
Number of tubes	384
Length of tubes	15 ft.
Water heating surface	2,498 sq. ft.
Superheating surface	895 "
Inclined grate area of Roney stoker	52 "

Results of Tests.

Date.	April 15, 1898.	April 26, 1898.
Duration	24 hours.	24 hours.
Kind of coal	George's Creek	Cumberland.
Average steam pressure,	107.9 lb.	107.6 lb.
Force of draft in flue	0.50 in.	1.28 in.
Temperature of feed water	39.0 degrees	39.6 degrees
" " escaping gases	742 degrees	695 degrees
Per cent. of CO_2 in gases	11.0%	8.3%
Dry coal consumed	25,972 lb.	47,866 lb.
" " " per hour	1,082 "	1,994 "
Per cent. of refuse in ash pit	9.7%	7.2%
Water evaporated per hour from and at 212 degrees	11,700 lb.	19,386 lb.
Water evaporated per pound of dry coal from and at 212 degrees	10.81 "	9.72 "
Water evaporated per pound of combustible from and at 212 degrees	11.98 "	10.48 "
Rating of boiler	250 h. p.	250 h. p.
Commercial horse-power	339 "	562 "
Per cent. over rating	36%	125%
Horse-power per square foot of grate	6.25 h. p.	10.81 h. p.
Water heating surface per horse-power	7.37 sq. ft.	4.44 sq. ft.
Dry coal burned per square foot of grate per hour	20.81 lb.	38.34 lb.
Dry coal burned per square foot of water heating surface per hour	0.43 "	0.78 "
Water evaporated per square foot of water heating surface per hour from and at 212 degrees	4.68 "	7.76 "

The results of the test of April 15 are about the same as for the test at the Bristol Manufacturing Company's, but that on April 26 is phenomenal. The economy is not good, but this is evidently due to the firing, as the CO_2 in the flue gases was low. Whether the firing could have been better, or whether the poor combustion was due to inexperience when forcing, I do not know. The result 4.44 square feet of water heating surface per horse-power is extremely small, and the coal consumption per square foot of grate, the coal burned per square foot of heating surface, and the water evaporated per square foot of heating surface are remarkably large. The percentage of power above the rating is no less noticeable. The temperature of the gases is low considering the coal consumed, although it would no doubt have been higher if the combustion had been better.

The Manning boiler has a very small water surface, and the ebullition during this test must have been tremendous. Nevertheless the steam left the boiler superheated 50 degrees, that is to say, if moisture was in the steam as it escaped from the water surface, it was, by contact with the tubes, evaporated and superheated.

Corliss Vertical Boiler at the Narragansett Mills.

On November 16 and 17, 1894, I made tests of Corliss vertical boilers at the Narragansett Mills, Fall River, Mass., with the results given below:

General Dimensions.

Diameter of shell	84 in.
Diameter of fire box	78¼ "
Height of fire box	5 ft.
Number of tubes	268
Diameter of tubes	2¼ in.
Length of tubes	13 ft.
Grate surface	33.4 sq. ft.
Water heating surface	1,492 "
Superheating surface	544 "
Grates	fixed and hand fired.

Results of the Tests.

Date.	November 16.	November 17.
Duration	10.87 hours.	6.06 hours.
Kind of coal	Pocahontas	
Calorific value of coal by analysis	15,036 B. t. u.	
Average steam pressure	115.93 lb.	114.55 lb.
Force of draft in smoke box	0.16 in.	0.19 in.
Temperature of feed water	106 degrees	103 degrees
" " escaping gases	501 "	555 "
Superheat	39 "	54 "
Dry coal consumed	7,253 lb.	4,241 lb.
Per cent. of refuse in ash pit	5.89%	7.45%
Water evaporated per hour from and at 212 degrees	7,073 lb.	7,370 lb.
Water evaporated per pound of dry coal from and at 212 degrees	10.60 "	10.43 "
Water evaporated per pound of combustible from and at 212 degrees	11.27 "	11.27 "
Commercial horse-power rated	160 h. p.	160 h. p.
" " on test	205 "	214 "
Excess of horse-power above rating	28%	34%
Horse-power per square foot of grate	6.14 h. p.	6.41 h. p.

Date.	November 16.	November 17.
Water heating surface per horse-power	7.28 sq. ft.	6.97 sq. ft.
Dry coal burned per square foot of grate per hour	20.00 lb.	21.17 lb.
Dry coal burned per square foot of water heating surface per hour	0.45 "	0.43 "
Water evaporated per square foot of water heating surface per hour from and at 212 degrees	4.66 "	4.83 "

Gunboat Boiler at the Washington Mills, Lawrence, Mass.

This is a type of horizontal fire tube boiler that in the Navy is called the Gunboat type. It consists of two cylindrical furnaces joining a combustion chamber, from which tubes pass to a smoke box at the farther end. There are four of these boilers at the Washington Mills. Two of them were tested simultaneously by me and the general results are given below as applied to one:

Diameter of boiler	10 ft.
" " furnaces	48 in.
" " tubes	2½ "
Length " "	14 ft.
Number " "	570
Heating surface, fire sides	5,300 sq. ft.
Grate area	60 "

Results of Test.

Date.	December 22, 1897.
Duration	11.5 hours.
Kind of coal	Pocahontas
Average steam pressure	122.3 lb.
Force of draft in smoke box	0.875 in.
Temperature of feed water	36.7 degrees
" " escaping gases	526 "
Dry coal consumed	21,842 lb.
Per cent. of refuse in ash pit	9.28%
Total water evaporated from and at 212 degrees	247,173 lb.
Water evaporated per pound of dry coal from and at 212 degrees	11.32 "
Water evaporated per pound of combustible from and at 212 degrees	12.47 "
Commercial horse-power	623 h.-p.
Horse-power per square foot of grate	10.38 "
Heating surface per horse-power	8.51 sq. ft.
Dry coal burned per square foot of grate per hour	31.66 lb.
" " " " " " heating surface per hour	0.36 "
Water evaporated per square foot of heating surface per hour from and at 212 degrees	4.05 "

This is an instance of a high rate of combustion, and fairly high rate of evaporation per square foot of heating surface. These boilers were designed with a large heating surface in proportion to the grate surface (88 to 1), and the rate of evaporation was correspondingly low.

Boilers of Locomotives.

To pursue this subject a little farther, the greatest forcing in boiler performance is being performed constantly by the fire tube boilers of locomotives. It is common enough for a horse-power to be obtained from two square feet of heating surface, or even less, and the horse-power per square foot of grate may vary from 20 to 40. In comparison with such performances, others are insignificant.

Conclusion.

In the data here presented, there are, I believe, ample reasons for believing that the fire tube type of boiler is fully equal to any demand that may be made upon it, and I am not aware of any evidence that the water tube type of boiler can surpass it in the respect which forms the subject of this paper.

DISCUSSION.

Prof. Wm. Kent.—The widespread impression which Mr. Dean says exists to the effect that the fire tube type of boiler has less capacity to be worked above its rated power than boilers of the water tube type is one that does not exist among persons who are well acquainted with both types as they are built to-day. Perhaps twenty-five years or more ago that impression was justified when fire tube boilers were built with tubes of small diameter crowded closely together and gas passages were so badly designed that they facilitated the passage of gas through some of the tubes to the neglect of others, and when some water tube boilers, now no longer in the market, were built with large grate surface and small heating surface. The impression gained from the comparison of such boilers found its way into print and may have been perpetuated in print ever since, but it is no longer true and it is not the "prevailing opinion" among those who have given some attention to the subject.

Mr. Dean gives the results of tests showing ratios of evaporation ranging from 4.30 to 7.76 pounds of water from and at 212 degrees per hour per square foot of heating surface. These figures are by no means extraordinary. I have obtained as high as 7.94 pounds with the old-fashioned two flue boiler; 6½ pounds with the old style water tube boiler at a ratio of heating to grate surface of 22 to 1 and discharge of gases into the chimney at about 900 degrees F.

The Babcock & Wilcox boiler showed as high as 13.67 in a test made by the Navy Department Board of one of the boilers built for the United States Cruiser "Cincinnati," as shown in my book on "Steam Boiler Economy," page 400. The Naval Board that tested the Hohenstein Marine Boiler obtained with Pocahontas coal an evaporation as high as 14.15 pounds from and at 212 degrees per hour per square foot of heating surface and with oil fuel they obtained as high as 16.7 pounds, which I think is the highest figure on record.

In fact almost any boiler can be driven at three times its normal rating or three times as hard as it should be driven in order to obtain its maximum economy, provided the facilities exist for burning a sufficient quantity of coal under it.

The conditions under which a boiler can be forced depend, not on the boiler itself, but on something apart from the boiler—considering the boiler to be an apparatus for absorbing heat and not for generating it. These conditions are: quality of coal, kind of grate, area of grate surface and volume of air supply. Given a free burning coal, shaking grates, large grate surface in proportion to the heating surface and forced draft and almost any boiler can be driven at such a rate as to cause the gases of combustion to leave the boiler at a thousand degrees or upwards and thus show high capacity at the expense of economy.

Prof. R. C. Carpenter.—The paper by Mr. Dean is one of no little interest from the fact that it collates a number of tests which are individually and collectively valuable. Mr. Dean states that there is a widespread impression among many persons that the fire-tube type of boiler has less capacity to be worked above its rated power than boilers of the water-tube type, and his tests are intended to show that such an impression is erroneous.

I am inclined to believe that if such an impression exists it is erroneous, but I have had the good fortune to be thrown in contact with a large number of persons interested in boilers, and I

have never found that such an impression was in existence. I do not wish to detract from the value of the paper by Mr. Dean, and I am very willing to admit that he has pounded his straw man very hard; but I really believe that the object of his paper should be changed, since, so far as my experience goes, the general impression held by interested people is that the fire-tube boilers are capable of being forced more than the water-tube.

Prof. E. A. Hitchcock.—In our locomotive testing it has always been a surprise to me that such high capacity, as well as good economy, was obtained from the locomotive boiler. For example, take these figures:

	Consolidation Freight.			
Heating surface	1880	1880	1880	1880
H. P. per sq. ft. heating surface	3.	3.1	3.3	2.9
Efficiency	59.4	63.6	59.6	65.5
	Passenger.			
Heating surface	1946	1946	1974	1974
H. P. per sq. ft. heating surface	4.3	3.6	4.	3.9
Efficiency	57.2	59 8	63.4	58.2

I have heard stated that the principal reason for the high steaming capacity of the locomotive was due to the fact of its constant vibration, and that the same boiler used stationary under the same conditions as to draft, coal, etc., would not give the same results. Does any member of the Society know this to be a fact?

On page 100, under "Results of the Tests" of a Corliss vertical boiler, Mr. Dean gives the calorific value of Pocahontas coal as 13,952 British thermal units. Is not that an error? Many Mahler calorimeter determinations by Professor Lord of the Ohio State University show that coal to give about 15,000 British thermal units.

Mr. A. H. Eldredge.—The paper presented by Mr. Dean is one of interest and of great importance to the engineering profession of the day.

In determining the kind of boiler to be used there are many questions to be considered, among which are the following:

1. The pressure to be carried.
2. The size of boiler and thickness of shell.
3. The probable time before the reduction of pressure by the

insurance companies will put the boiler out of commission, or at least, out of range of economical usefulness.

4. The forcing capacity of the boiler.
5. The comparative safety of the different types of boilers.
6. Reliability for continuous hard service.
7. Size of the plant.

These are all questions that the engineer dealing with plants from 500 to 5,000 horse-power or over must meet. Where should the fire tube boiler stop and the water tube begin?

In dealing with some of the largest fire tube boiler companies during the past year they have quoted prices based on 14 square feet of heating surface, and when in competition with the water tube builders they are willing to quote on 12 square feet. The writer has been unable to get the fire tube builders to guarantee either evaporation or repairs, but has found the water tube builders willing to do either.

Referring to the 24 year old boiler of the Atlantic Mills, the writer would ask if this boiler was used 24 hours a day under the conditions stated. The burning of 21.4 pounds of coal per square foot of grate with 0.21-inch draft was an excellent performance. During the past summer the writer had one plant of 11 fire tube boilers averaging 150 horse-power each, and with 0.295-inch draft it was impossible to burn over 9¼ pounds of New River coal per square foot of grate as the average for 24 hours, the boilers being pushed for the 24 hours.

Forcing Capacity of Water Tube Boilers.

In 1901, on a plant of nearly 3,000 horse-power of water tube boiler, the writer had 75 tests made to determine the comparative values of different western fuels. One boiler was fitted up with scales, tanks, pump, etc., for the work, and was handled exactly as the rest of the battery. In only three tests, with the poorest coals, did the boiler simply reach rated horse-power, while in most tests it ran from 25 to 60 per cent. over capacity, and in a few cases ran over 85 per cent. above rating. These boilers are now eight years old and have been pushed this way day and night with poor feed water, and are still carrying 150 pounds pressure, with every indication that they will continue to do so.

Is there not a best place for each type of boiler?

Mr. A. Bement.—The author performs a useful service in

directing attention to the injustice done the horizontal return tubular boiler by present methods of rating its capacity. This style of boiler is largely made in local shops and not by enterprising manufacturing companies. For this reason few people have presented arguments in its favor, or taken the trouble to ascertain its value, therefore, its more enterprising rival, the water tube boiler, has gone ahead of it, and secured a commercial advantage.

The capacity produced by a boiler is dependent on two things: 1. Quantity and disposition of heating surface. 2. Amount of heat supplied to it. The latter has much more influence than the former, because if a boiler is supplied with a large quantity of heat, it will produce a large capacity. Decreasing efficiency, of course, reduces the capacity, but at a very much lower rate than the increase due to increasing quantity of heat supplied, therefore large capacity is more dependent on the fire than on the boiler.

A number of years since on the Northern Railway of France, they divided a locomotive boiler into different portions, so that the evaporation taking place in each could be measured. One of these divisions was the firebox only, from which the evaporation was stated to be as high as 35 pounds per square foot of heating surface. It is probable that the forcing capacity of many boilers, including both fire and water tube types, is very often underestimated.

The limits of forcing are reached when the metal can no longer transmit the heat without damage, and when the boiler begins to discharge water with the steam; but as the ability of the metal to withstand heat is dependent on the presence of water in contact with it, it follows that the limit is governed by circulation and the ability of the boiler to make dry steam. In a number of cases I have found that this latter feature limited the horse power to a much lower quantity than would have been the case if the dry steaming capacity of the boilers had been in proportion to their circulating capacity.

I do not think that the author's figures indicate the limits of forcing capacity of fire tube boilers, and assuming circulating and dry steaming capacity the same for water tube as for fire tube boilers, I think each possesses the same capacity for output, but efficiency as well as capacity is of importance and it is my opinion that the efficiency of the average return tubular as now

made, is higher than the majority of water tube boilers now in service, but not higher than that of the best water tube boilers, and in an endeavor to illustrate this feature, the figures in the following table are offered:

	Return Tubular. S. D. Warren.	B. & W. 12 tubes high.	Heine. A	Heine. A	Heine. B	Atlantic Mills. 1st Test.
Square feet of heating surface per horse-power.....	5.43	5.96	5.68	8.5	8.3	8.0
Temperature of escaping gases above steam temperature.....	311	261	476	278	92	46
CO_2	14.35	11.1	11.5	10.7	10.3	Not given.

The boilers designated as A. & B. are those referred to in a paper presented by me on the present occasion, boiler B having had the original travel of the gases changed, so that they flowed in contact with a much greater portion of the tube surface than formerly. From the above comparison, it would appear that the Warren boiler is not as efficient as the Heine B and not much different from the Babcock & Wilcox but much better than the Heine A. The Atlantic Mills boiler, however, shows an efficiency at least as high as the Improved Heine B, if it can be assumed that the temperature of escaping gases as given is correct. Regarding this there is some doubt in my mind, because it may be that the location of the measuring instrument was not such as to show the average temperature of the flowing gases, but if the figure as given may be accepted, it appears that its temperature of escaping gases is considerably lower than with the Heine B, and its capacity is also larger; but on the other hand, it is probable that the condition of combustion was better, which would have an important effect. It may be that this Atlantic Mills boiler, owing to its proportions, had more of its surface acted on by the hot gases than with the Warren boiler, and it is not certain that the gases flow equally in contact with the tube surface of all fire tube boilers, and it is reasonable to expect that they do not, in view of their failure to do so to a serious extent with many water tube boilers.

The foregoing remarks refer to the return tubular boiler as usually set, having the fire underneath, but if set with one end to the furnace, so that the heat does not act on the shell, but only on one end, and the tube surface, its capacity will be limited by

the durability of the joint of the tube ends with the head, on account of the effect of high temperature on these joints, causing leakage. Similar limit to capacity occurs with water tube boilers having the lower row of tubes exposed for their full length to the heat of the furnace, especially in those cases where the tube surface above the bottom row is removed from the direct effect of the fire by tiling.

As far as circulation, or, in other words, water supply to the heating surface and durability of the heating surface is concerned, it is probable that the return tubular boiler has a greater forcing capacity than have the majority of water tube boilers, but not greater than possible with a well designed one.

Mr. James H. Abercrombie.—Par. 5. Definition of Forcing Capacity.

I fear comparisons made on the basis of the definition given may be misleading because there is no universally accepted rule for measurement of heating surface, or for rating any given type of boiler; and the practice of makers in these matters no doubt varies according to individual views. Some of the vagueness which necessarily surrounds any such comparison might be removed, I think, if statements of rating were required to include statements of the assumed rate of evaporation per square foot of heating surface, of the method of measurement (whether fire surfaces or water surfaces) and also of the capacity per unit area of floor space.

I believe Mr. Dean is conservative in his contention that the horizontal tubular boiler is at least equal to the water tube in its capacity per square foot of heating surface, and nothing would please me better than to have him show a similar equality, or a superiority, in capacity referred to floor space occupied.

I have not studied this aspect of the matter closely enough to justify positive general statements, and not at all in connection with boilers 90 inches in diameter; but a comparison of several types of water tube boilers with 68-inch horizontal return tubular boilers for conditions requiring economy of floor space did not deter me from selecting the latter for new installations.

In this connection it is worth noting that the spaces required between the settings of water tube boilers for cleaning purposes operate to reduce the capacity of these boilers per unit area of floor space.

Pars. 6 to 9. In making comparisons of various boilers a few years ago, I found the following water capacities:

Horizontal water tube,	from	8.4	to	10.6	lb.	per	sq.	ft.	of	heating surface.
Vertical	"	"	"	7.2	"	10.9	"	"	"	"	"	"
Horizontal return tubular	"	9.3	"	13.4	"	"	"	"	"	"

The figures for the water tube boilers were furnished by the makers; the others were estimated. The largest horizontal tubular boiler considered was 72 inches diameter with 9.3 lb., and a 64-inch and 68-inch boiler of somewhat unusual proportions with 13.4 and 13 pounds respectively are included.

The amount of variation shown in these figures for the horizontal tubular boilers might be expected. The addition of tubes to a water tube boiler adds both to heating surface and water capacity and will not affect the ratio much; but the addition of tubes to a horizontal return tubular boiler adds to the heating surface and *takes from* the water capacity. The height of water line also may affect the ratio considerably. In view of what these figures show I think Mr. Dean should rely rather on his argument in paragraph 9, showing that quantity of water is of minor importance so far as forcing capacity is concerned, than on his statistics, to prove this part of his case. Other things being equal, I would give preference to the boiler containing the larger quantity of water, and therefore I believe that a horizontal tubular boiler containing the largest practicable number of tubes is not the best representative of its type. These boilers can have fewer tubes than are generally fitted, with little loss of capacity or efficiency. An extreme case will illustrate this. I have found by trial that the stopping of 60 out of 136 tubes in a 72-inch boiler, while using the same kind and quantity of coal per hour, reduced the capacity 12 per cent. and the evaporation per pound of coal 13.4 per cent. only, although the total heating surface was reduced 39 per cent., and the tube surface 44 per cent. In the test with all tubes in use the rate of evaporation worked out at 2.95 pounds, and with 60 tubes stopped 4.18 pounds per square foot of heating surface per hour. Note that these two rates of evaporation were obtained with the same fire conditions, the increase being due simply to cutting out a considerable amount of comparatively inefficient heating surface. This leads me to think that if it is desired, in connection with the subject of forcing capacity, to consider the possibility of injury to boilers, the rate of evaporation per square foot of heating surface should be ignored, and attention given only to the intensity and local effect of combustion.

Par. 10 and 11. I can see nothing wrong in the criticism of the claims referred to as it stands, but I think this criticism naturally calls attention to the possibility, in horizontal return tubular boilers, of having so much tube area that the gases select the upper rows of tubes and are comparatively sluggish in the lower ones so that these latter may be no more efficient than the water tubes subject to the conditions criticised.

An advantage incidental to the use of fewer tubes than usual is that it permits of a lower water line, giving a larger area for releasing steam, and a slower drop of water line when steaming hard, although it may reduce slightly the shell heating surface. These points have some relation, perhaps an important one, to forcing capacity.

Experience with locomotive and marine boilers would seem to show that considerable advance may be made in forcing stationary boilers beyond what is average practice without incurring risk of injury to boilers which are well cared for.

In the case of boilers using small anthracite fuel, the limit of forcing will usually be set by the grate area, and this may apply to other fuels of low heating power.

Where good bituminous coal is used the limit may *still* be set by grate area, but in some cases within that limit priming may be set up, and also proper regulation of the water level may be difficult.

I heartily agree with Mr. Dean's concluding statement, and as having some bearing on this whole subject I offer the following brief description of two almost identical plants designed about three years ago, the basis of the design being the belief that horizontal return tubular boilers could be arranged to give as much capacity per sqare foot of boiler room floor occupied, as water tube boilers could give, and also that aside from equal capacity there would be at *least equal* economy of evaporation, and *superior* economy in the matter of cleaning and repairs.

In arranging these plants it was endeavored by the liberal use of economizers to confine the work of the boilers as much as possible to evaporating, imposing on them as little heating as possible, thus securing a high rate of evaporation per square foot of heating surface with a comparatively moderate rate of transmission of heat through the plates.

Each plant is made up of 5 units, and each unit consists of two horizontal return tubular boilers, and one Green fuel economizer.

Each boiler is 68 inches diameter by 16 feet 11½ inches long, and has 86 3-inch tubes. The economizer has 240 pipes. The two boilers are suspended over one fire, the grate being 6 feet deep and 12 feet wide, and they deliver gases into a common uptake. The economizer is set immediately over the firing floor. It is in two sections, set side by side. One section which is 6 pipes wide receives the gases through a short uptake. The gases pass forward through this wide section and return through another section which is 4 pipes wide, and then pass through a short connection to a main flue above the boilers, leading to induced draught fans. The setting walls are built with 10-inch spaces between, and through these spaces the warmed air above the boilers is drawn down to the grates, the ashpit doors being kept closed.

The boilers were intended to evaporate, if required, 5 pounds of water per square foot of boiler heating surface and have done more at times, although the usual requirements do not demand quite that rate of evaporation. The usual fuel is Wyoming Rice coal (No. 2 Buckwheat). With bituminous coal the designed evaporation is easy, while with the Rice it is probably near the comfortable limit for steady general work, although it has been considerably exceeded on individual units.

A few tests were made with the object of roughly comparing the performance of a 72-inch horizontal tubular boiler having 136 3-inch tubes, and without an economizer, with that of one unit (two boilers and one economizer) of the new plant. Rice coal and also bituminous coal were used. With Rice coal the apparent actual evaporation per square foot of heating surface was 1.96 pounds in the 72-inch boiler and 5.19 pounds in the new unit (boiler surface only). The evaporation at and from 212 degrees was 8.494 pounds and 9.31 pounds respectively.

In the case of the 72-inch boiler, the water was fed at 127.5 degrees and the gases left at 487 degrees. In the case of the new unit the water entered the economizer at 162.5 degrees and the boilers at 292 degrees; and the gases left the boilers at 535 degrees, and left the economizer at 279 degrees.

With bituminous coal the apparent actual evaporation per square foot of heating surface was 2.15 pounds in the 72-inch boiler, and 6.36 pounds in the new unit (boiler surface only). The evaporation from and at 212 degrees was 10.3 pounds and 11.3 pounds respectively. In the case of the 72-inch boiler the water was fed at 126.3 degrees, and the gases left at 478 degrees.

In the case of the new unit the water entered the economizer at 135.4 degrees, and the boilers at 256 degrees; and gases left the boilers at 582 degrees, and left the economizer at 269 degrees.

Thus, at a rate of evaporation per square foot of heating surface (boilers only) per hour of 2.6 to 1 in the case of Rice coal, and nearly 3 to 1 in the case of bituminous coal the new unit showed a greater evaporation per pound of coal to the extent of about 9.6 per cent. The pressure in the case of the 72-inch boiler was about 85 pounds and in the case of the new unit about 95 pounds. The coal supplied to both outfits was from the same lot in each case.

Assuming the figures to be accurate as they stand, without correction for moisture in coal or in steam, it would appear that the economy of the boilers alone in the new unit is only a trifle inferior to that of the 72-inch boiler at its low rating, the heat added by the economizer being nearly all clear gain.

To give another illustration of the capacity of these new units, I may state that 4 out of 5 in one battery using Rice coal have quite comfortably supplied steam for 2,455 horse power of compound Corliss condensing engines, 96 horse-power simple Corliss engine, about 55 horse-power of small slide valve engines, and about 35 horse-power of duplex pumps.

Also these 5 units are doing the work which was previously done by ten 66-inch by 16 feet horizontal tubular boilers, and six horizontal water tube boilers rated at 208 horse-power each, with two detached economizers of 320 pipes each, and doing it more easily and economically and on less than half the floor space, although with 10 feet higher roof. Length of floor covered is 75 feet.

While on this subject, it occurs to me to suggest that shell surface being evidently the best part of the heating surface in a tubular boiler, a very good combination might be such as I have described with a Cornish or Lancashire boiler substituted for the multitubular. I think with that arrangement a very high rate of evaporation per square foot of heating surface would be obtainable without improper forcing.

*Mr. F. W. Dean.**—I have read with interest the discussions of my paper, and am gratified to see that there is general agreement with my claim as to the forcing capacity of fire tube boilers.

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* Author's Closure, under the Rules.

In regard to Professor Carpenter's claim that I have erected a straw man to pound, I have obtained my ideas on the subject from hearing the comments of many men, and I believe that it is commonly stated by the agents of water tube boilers that they will permit forcing to a greater extent than fire tube boilers. Professor Kent states that he thinks that this idea once existed but does no longer. I hope that that is true.

In Mr. Kent's discussion he states that any boiler can be forced to any extent if sufficient coal is burned, and that I state at the end of paragraph 5 in my paper.

Mr. Bement gives some temperatures of escaping gases above the temperatures of the steam, and draws some conclusions with reference thereto. In the case of the Warren boilers, the brickwork was in excellent condition, and the temperatures given are probably more nearly correct than occurs in tests of many boilers. In the case of water tube boilers, there are so many cleaning doors and large doors at the ends that there is likely to be a much greater leakage of air into the fire, and thereby a greater reduction of the temperature of escaping gases due to that cause than occurs in fire tube boilers, and I should not be surprised if some of the temperatures that he gives are too low. We had strong evidence of this in the case of some water tube boilers alongside of the Warren tubular boilers. While both kinds of boiler had the same grate and were fired by the same men, it was always found that the escaping gases from the water tube boilers had much more free oxygen than the horizontal tubular boilers, and it was found impossible to stop the leaks around the cleaning and front doors sufficiently to equalize the boilers in this respect. If the joints around the cleaning door frames are cemented they will very soon expand and contract sufficiently to push the brickwork still farther away so that the original condition again occurs.

Professor Hitchcock asks if there is not an error in the calorific value of the Pocahontas coal, given in the results of the tests of the Corliss boiler at the Narragansett mills. I find that there is an error and have corrected it in the paper.

No. 1059.*

PRESSURES AND TEMPERATURES IN FREE EXPANSION.

BY A. BORSODY AND R. CAMPBELL CAIRNCROSS.
(Non-Members.)

PRESENTED BY C. E. LUCKE, NEW YORK CITY.
(Associate Member of the Society.)

1. The following series of pressures and temperatures presented in tabular and graphic form were obtained experimentally in the laboratory of Columbia University in the spring of 1904. The work is part of a thesis problem, the text of which is not of general interest, but the curves, it is believed, are worthy of some discussion. It is for the purpose of stimulating a free discussion that these experimental results are presented by the authors.

2. The pressure at any point of a nozzle has usually been measured by the introduction of a small searching tube connected to a pressure recorder; but as the steam or air is in rapid motion all pressures so measured must be corrected for velocity effects to get true pressures. This is troublesome, as it involves three unknown quantities, the true velocity, the density, and eddy current effects. To get some data on these effects, three methods of introducing the searching tube were used, one with the end of the tube opposed to the current, one with the end of the tube in the direction of the current, and the third with the opening in the side of the tube. Thus three curves of pressures are reported for each case. The searching tube was passed through a stuffing box prepared by using an angle valve and breaking off the valve proper, and the external part of the tube was fastened to a scale of meters while its interior was connected to a corrected pressure gauge for high pressures and a mercury column for low ones. The drawing will explain this detail, together with the steam chamber for feeding the nozzle, which it will be seen was constructed of pipe fittings.

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

3. Temperatures in the nozzle when measured by thermometers are incorrect because of the heat set free on impact of the swiftly moving gases. Similarly a thermal couple across the jet will also read high; for this reason the couple was made of thin nickel copper wire, smooth as possible, and stretched through the nozzle. The curve of calibration of the couple is given and the electrical connections for reading temperatures will be clear from the drawing.

4. The rest of the paper reports the results of these pressure temperature measurements for steam and expanding air.

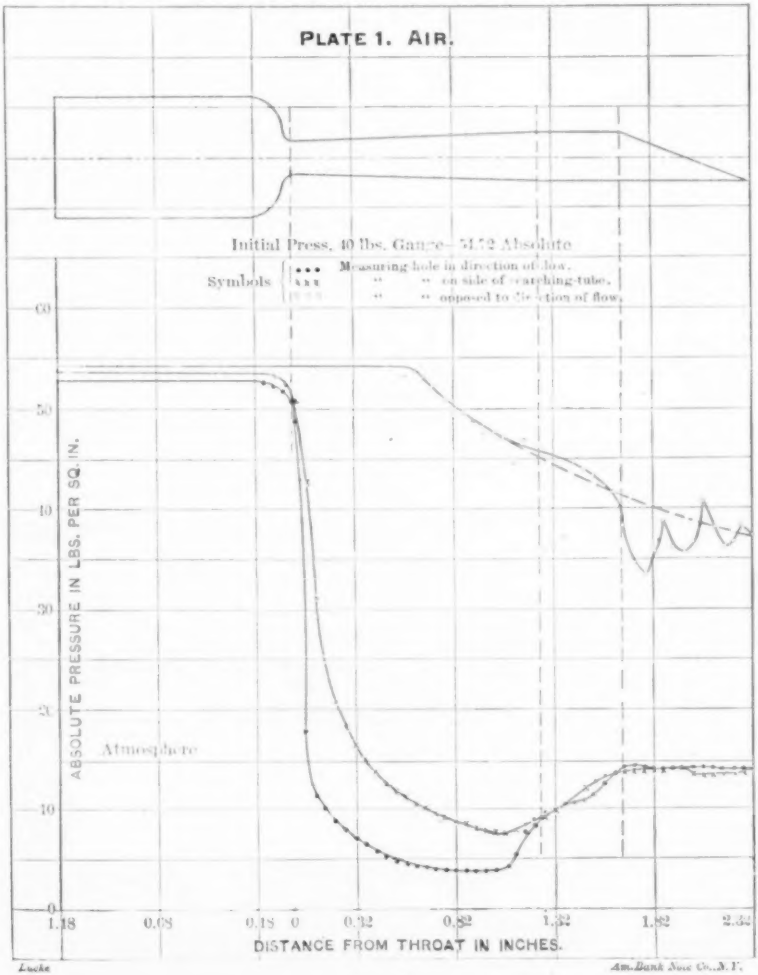

FIG. 8.

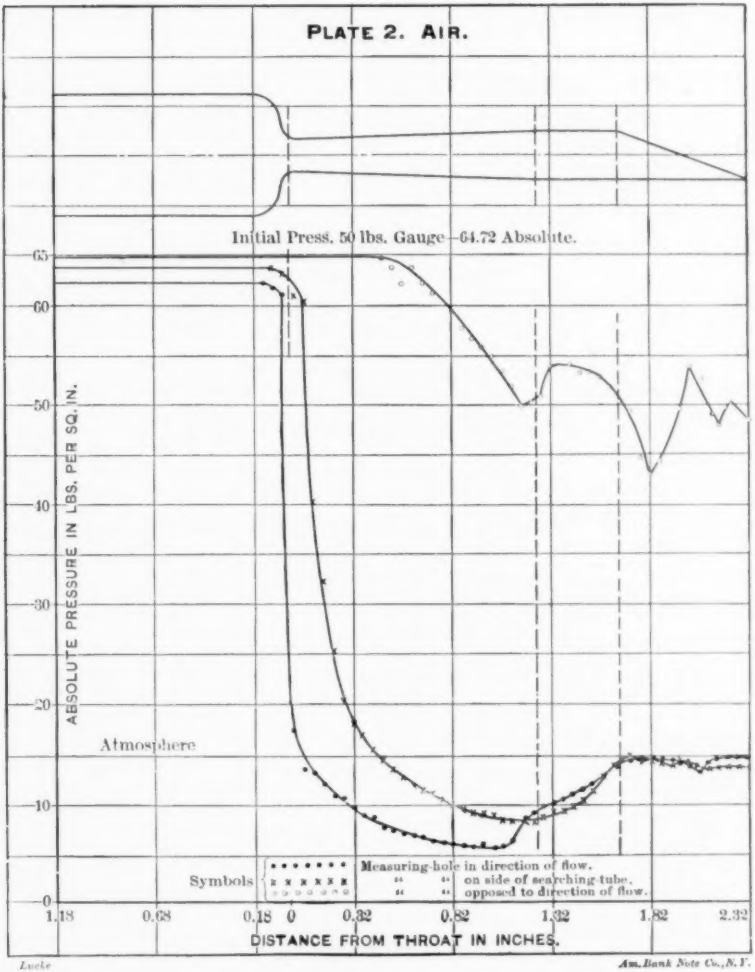
PLATE 2. AIR.
Initial Press. 50 lbs. Gauge—64.72 Absolute.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
65
60
50
40
30
20
10
0
Atmosphere
Symbols
Measuring-hole in direction of flow.
" " on side of searching tube.
" " opposed to direction of flow.
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 9.

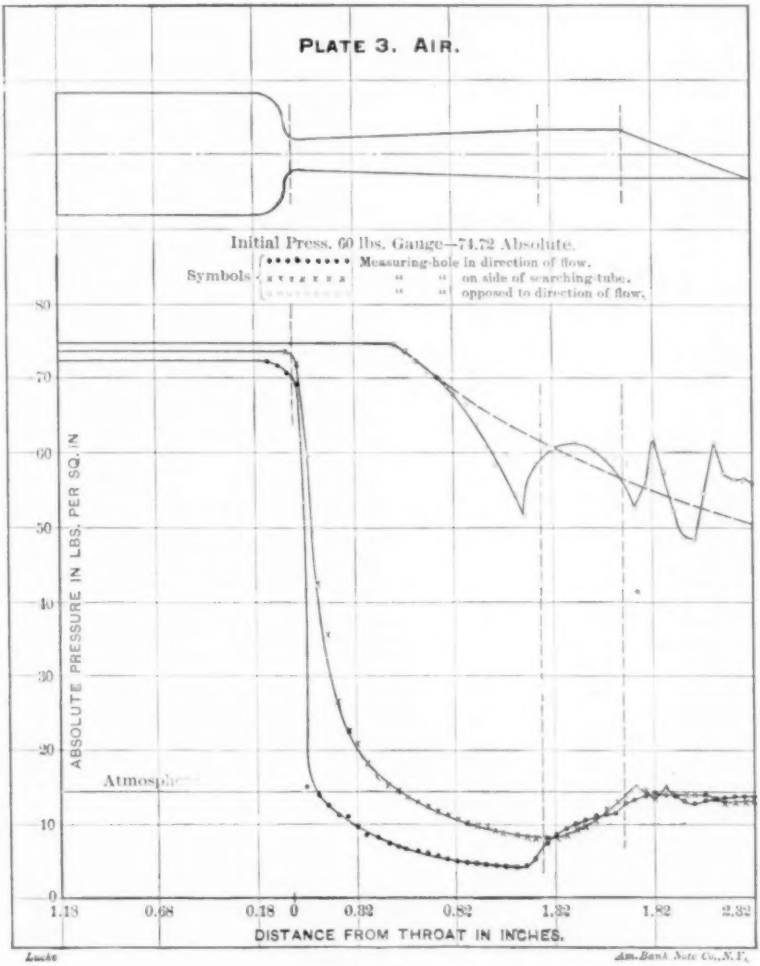
PLATE 3. AIR.
Initial Press. 60 lbs. Gauge—74.72 Absolute.
Symbols
Measuring-hole in direction of flow.
" " on side of searching-tube.
" " opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 10.

AIR—INITIAL PRESSURE (40 POUND GAUGE — 54.72 ABSOLUTE)

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in
−0.53	.....	52.72	.....	53.47	.....	54.22
−0.48	.....	52.72	.....	53.47	.....	54.22
−0.43	.....	52.72	.....	53.47	.....	54.22
−0.38	.....	52.72	.....	53.47	.....	54.22
−0.33	.....	52.72	.....	53.47	.....	54.22
−0.28	.....	52.72	.....	53.47	.....	54.22
−0.23	.....	52.72	.....	53.47	.....	54.22
−0.18	.....	52.72	.....	53.47	.....	54.22
−0.13	.....	52.72	.....	53.47	.....	54.22
−0.08	.....	52.22	.....	53.47	.....	54.22
−0.03	.....	51.72	.....	52.47	.....	54.22
+0.02	.....	49.22	.....	50.47	.....	54.22
0.07	36.25	17.76	.....	42.72	.....	54.22
0.12	22.95	11.23	.....	31.22	.....	54.22
0.17	20.55	10.06	.....	24.72	.....	54.22
0.22	18.35	8.98	42.45	20.81	.....	54.22
0.27	16.65	8.14	37.05	18.16	.....	54.22
0.32	15.45	7.55	33.85	16.59	.....	54.22
0.37	13.75	6.72	29.85	14.82	.....	54.22
0.42	12.35	6.03	27.85	13.64	.....	54.22
0.47	10.75	5.24	25.85	12.66	.....	54.22
0.52	9.75	4.75	24.25	11.87	.....	54.22
0.57	9.35	4.56	23.25	11.38	.....	53.22
0.62	9.05	4.41	21.85	10.69	.....	53.72
0.67	9.05	4.41	20.55	10.06	.....	53.22
0.72	8.45	4.11	19.85	9.71	.....	52.22
0.77	8.35	4.07	18.85	9.22	.....	50.72
0.82	7.95	3.87	18.05	8.83	.....	49.72
0.87	7.85	3.82	17.15	8.39	.....	49.72
0.92	7.85	3.82	16.35	7.99	.....	48.72
0.97	8.05	3.92	16.25	7.94	.....	47.72
1.02	8.15	3.97	15.95	7.80	.....	47.72
1.07	8.55	4.16	15.55	7.52	.....	47.22
1.12	10.75	5.24	16.15	7.90	.....	44.72
1.17	16.05	7.88	16.85	8.24	.....	46.72
1.22	18.85	9.22	17.85	8.73	.....	44.22
1.27	19.85	9.71	18.55	9.07	.....	35.72
1.32	20.35	9.96	19.85	9.71	.....	36.22
1.37	21.75	10.64	21.65	10.60	.....	37.22
1.42	22.25	10.89	22.25	10.89	.....	44.72
1.47	22.55	10.99	24.55	12.03	.....	43.72
1.52	23.55	11.53	26.25	12.85	.....	43.22
1.57	25.85	12.66	26.45	12.95	.....	42.22
1.62	27.55	13.49	27.05	13.25	.....	41.72
1.67	29.25	14.33	28.05	13.74	.....	38.22
1.72	29.45	14.43	28.35	13.89	.....	34.72
1.77	29.25	14.33	28.45	13.93	.....	33.72
1.82	28.75	14.08	27.85	13.64	.....	35.72
1.87	28.65	14.03	27.95	13.69	.....	38.72
1.92	28.55	13.98	27.65	13.54	.....	36.22
1.97	29.35	14.38	27.65	13.54	.....	35.72
2.02	29.35	14.38	27.55	13.49	.....	36.72
2.07	29.25	14.33	27.55	13.49	.....	40.72
2.12	29.25	14.33	27.65	13.54	.....	38.22
2.17	29.15	14.28	27.75	13.59	.....	36.22
2.22	29.05	14.23	27.95	13.69	.....	36.72
2.27	29.05	14.23	28.05	13.74	.....	38.22
2.32	28.85	14.13	28.25	13.84	.....	37.22

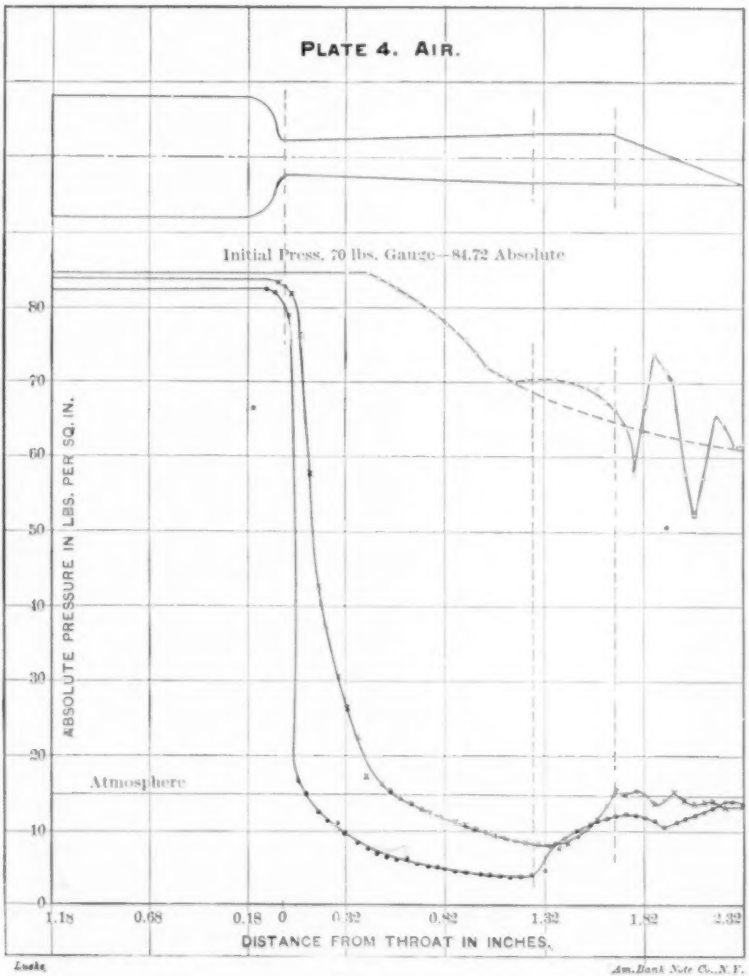
PLATE 4. AIR.
Initial Press. 70 lbs. Gauge—84.72 Absolute
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
Atmosphere
0
10
20
30
40
50
60
70
80
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 11.

PLATE 5. AIR.
Initial Press. 80 lbs. Gauge—94.72 Absolute
Symbols
Measuring-hole in direction of flow.
" " on side of searching-tube.
" " opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 12.

AIR—INITIAL PRESSURE (50-POUND GAUGE = 64.72 ABSOLUTE).

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	62.22	.....	63.72	.....	64.72
−0.43	.....	62.22	.....	63.72	.....	64.72
−0.38	.....	62.22	.....	63.72	.....	64.72
−0.33	.....	62.22	.....	63.72	.....	64.72
−0.28	.....	62.22	.....	63.72	.....	64.72
−0.23	.....	62.22	.....	63.72	.....	64.72
−0.18	.....	62.22	.....	63.72	.....	64.72
−0.13	.....	62.22	.....	63.72	.....	64.72
−0.08	.....	61.72	.....	63.72	.....	64.72
−0.03	.....	61.22	.....	63.22	.....	64.72
+0.02	35.45	17.37	.....	60.97	.....	64.72
0.07	27.35	13.39	.....	60.47	.....	64.72
0.12	27.25	13.35	.....	40.22	.....	64.72
0.17	24.45	11.97	.....	32.22	.....	64.72
0.22	21.95	10.74	.....	25.22	.....	64.72
0.27	21.45	10.50	.....	20.22	.....	64.72
0.32	19.45	9.52	36.85	18.06	.....	64.72
0.37	17.85	8.73	34.25	16.78	.....	64.72
0.42	16.25	7.94	30.95	15.16	.....	64.72
0.47	15.35	7.50	29.05	14.23	.....	64.72
0.52	14.95	7.31	27.65	13.54	.....	63.72
0.57	14.45	7.06	25.65	12.56	.....	62.22
0.62	13.85	6.77	24.25	11.87	.....	63.72
0.67	13.45	6.57	23.25	11.38	.....	62.22
0.72	13.05	6.37	22.25	10.89	.....	61.22
0.77	12.75	6.23	20.85	10.20	.....	59.72
0.82	12.35	6.03	20.05	9.81	.....	59.22
0.87	12.05	5.88	19.05	9.32	.....	57.72
0.92	11.85	5.78	18.45	9.03	.....	56.72
0.97	11.45	5.59	18.25	8.93	.....	55.72
1.02	11.35	5.54	17.55	8.58	.....	54.72
1.07	11.35	5.54	16.85	8.24	.....	53.22
1.12	12.25	5.98	16.55	8.09	.....	51.72
1.17	17.05	8.34	16.05	7.85	.....	49.72
1.22	17.95	8.78	16.25	7.94	.....	50.22
1.27	19.35	9.47	17.45	8.53	.....	50.72
1.32	20.05	9.81	18.25	8.93	.....	53.72
1.37	20.75	10.15	19.05	9.32	.....	53.22
1.42	21.65	10.60	19.85	9.71	.....	54.02
1.47	23.05	11.28	21.05	10.30	.....	53.22
1.52	23.65	11.58	22.35	10.94	.....	53.22
1.57	25.05	12.26	24.55	12.03	.....	52.72
1.62	26.85	13.15	27.65	13.54	.....	51.72
1.67	29.05	14.23	29.65	14.52	.....	50.22
1.72	29.45	14.43	29.05	14.23	.....	49.22
1.77	29.55	14.47	30.35	14.87	.....	44.72
1.82	29.45	14.43	28.45	13.93	.....	43.22
1.87	29.25	14.33	29.25	14.33	.....	44.22
1.92	29.25	14.33	28.65	14.03	.....	46.22
1.97	29.05	14.23	27.75	13.59	.....	49.72
2.02	28.05	13.74	28.45	13.93	.....	54.22
2.07	26.45	12.95	27.85	13.64	.....	52.72
2.12	29.45	14.43	27.75	13.59	.....	34.72
2.17	29.65	14.52	27.65	13.54	.....	48.22
2.22	29.65	14.52	27.55	13.49	.....	47.22
2.27	29.65	14.52	27.75	13.59	.....	49.22
2.32	29.55	14.47	27.85	13.64	.....	48.22

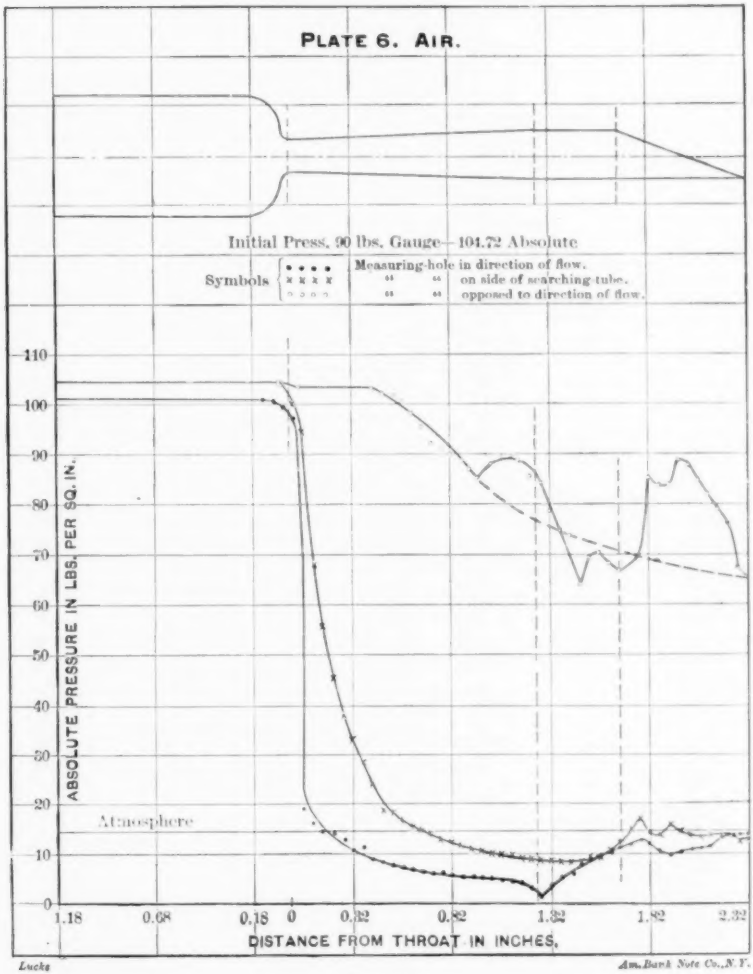
PLATE 6. AIR.
Initial Press. 90 lbs. Gauge—104.72 Absolute
Symbols
Measuring-hole in direction of flow.
on side of searching-tube.
opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
110
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 13.

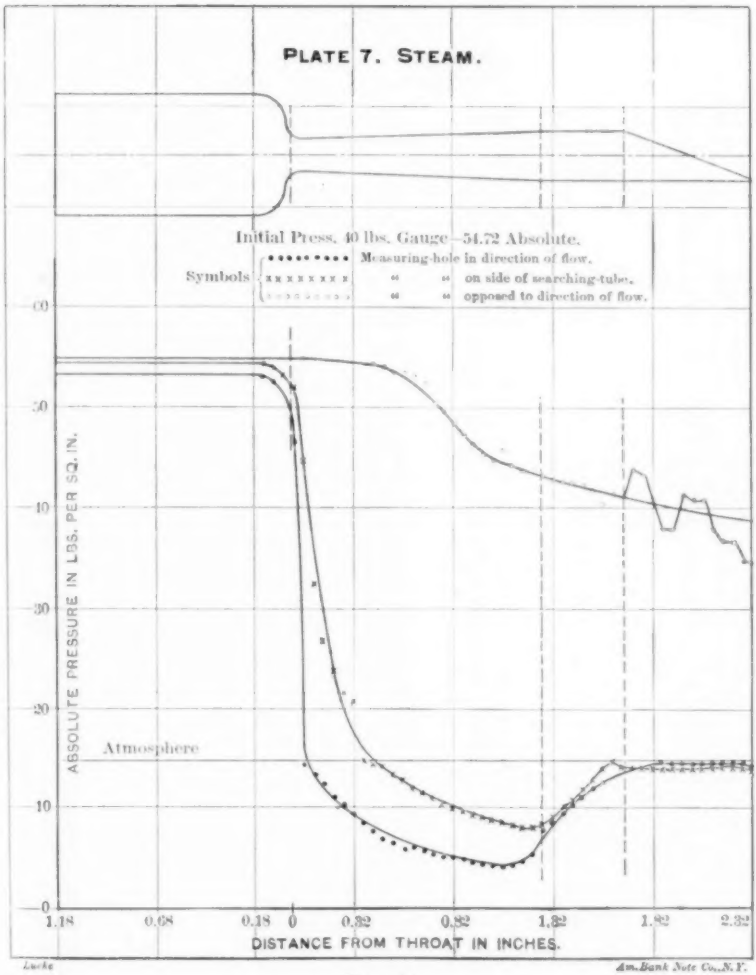
PLATE 7. STEAM.
Initial Press. 40 lbs. Gauge—54.72 Absolute.
Symbols
Measuring-hole in direction of flow.
on side of searching-tube.
opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 14.

AIR—INITIAL PRESSURE (60-POUND GAUGE — 74.72 ABSOLUTE).

DISTANCE FROM THROAT IN INCHES.	MEASURING HOLE IN THE DIRECTION OF FLOW.		MEASURING HOLE ON SIDE OF SEARCHING TUBE.		MEASURING HOLE OPPOSED TO DIRECTION OF FLOW.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.53	.....	72.22	.....	73.72	.....	74.22
−0.48	.....	72.22	.....	73.72	.....	74.22
−0.43	.....	72.22	.....	73.72	.....	74.22
−0.38	.....	72.22	.....	73.72	.....	74.22
−0.33	.....	72.22	.....	73.72	.....	74.22
−0.28	.....	72.22	.....	73.72	.....	74.22
−0.23	.....	72.22	.....	73.72	.....	74.22
−0.18	.....	72.22	.....	73.72	.....	74.22
−0.13	.....	72.22	.....	73.72	.....	74.22
−0.08	.....	71.72	.....	73.72	.....	74.22
−0.03	.....	70.72	.....	73.72	.....	74.22
+0.02	.....	69.22	.....	71.72	.....	74.22
0.07	30.55	14.97	.....	59.72	.....	74.22
0.12	29.35	14.38	.....	42.22	.....	74.22
0.17	25.05	12.26	.....	35.72	.....	74.22
0.22	22.45	10.99	.....	26.72	.....	74.22
0.27	22.35	10.94	.....	22.72	.....	74.22
0.32	20.05	9.81	.....	20.72	.....	74.22
0.37	17.55	8.58	36.85	18.06	.....	74.22
0.42	16.85	8.24	33.85	16.59	.....	74.22
0.47	14.55	7.11	31.55	15.46	.....	74.22
0.52	14.05	6.86	30.05	14.72	.....	74.22
0.57	13.45	6.57	28.05	13.74	.....	73.72
0.62	12.85	6.27	26.45	12.95	.....	72.22
0.67	12.55	6.13	25.05	12.26	.....	70.72
0.72	11.75	5.73	24.05	11.77	.....	70.22
0.77	11.15	5.44	22.25	10.89	.....	68.72
0.82	10.65	5.19	21.45	10.50	.....	67.72
0.87	10.35	5.05	20.35	9.96	.....	66.22
0.92	10.05	4.90	19.55	9.56	.....	63.72
0.97	9.75	4.75	19.25	9.42	.....	60.72
1.02	9.35	4.56	18.75	9.17	.....	58.72
1.07	8.95	4.36	18.05	8.83	.....	59.72
1.12	8.75	4.27	17.65	8.63	.....	54.22
1.17	9.25	4.51	17.05	8.34	.....	51.22
1.22	10.45	5.10	16.55	8.09	.....	58.72
1.27	16.05	7.85	16.05	7.85	.....	60.72
1.32	17.55	8.58	16.55	8.09	.....	60.22
1.37	19.05	9.32	17.95	8.78	.....	59.72
1.42	20.35	9.96	19.35	9.47	.....	61.72
1.47	21.45	10.50	20.45	10.01	.....	60.72
1.52	22.75	11.14	21.65	10.60	.....	60.22
1.57	23.25	11.38	24.35	11.92	.....	59.22
1.62	24.05	11.77	26.75	13.10	.....	59.22
1.67	27.25	13.35	29.05	14.23	.....	58.72
1.72	27.75	13.59	32.05	15.70	.....	52.72
1.77	28.25	13.84	29.75	14.57	.....	55.22
1.82	29.25	14.33	28.25	13.84	.....	61.22
1.87	29.05	14.23	31.55	15.46	.....	57.22
1.92	28.65	14.03	29.25	14.33	.....	52.22
1.97	26.65	13.05	27.55	13.49	.....	48.72
2.02	26.35	12.90	29.05	14.23	.....	48.22
2.07	27.35	13.39	29.05	14.23	.....	54.22
2.12	27.85	13.64	27.85	13.64	.....	61.22
2.17	28.65	14.03	28.25	13.84	.....	57.22
2.22	29.25	14.32	28.25	13.84	.....	56.22
2.27	29.65	14.52	27.65	13.54	.....	57.72
2.32	29.45	14.43	27.75	13.59	.....	55.22

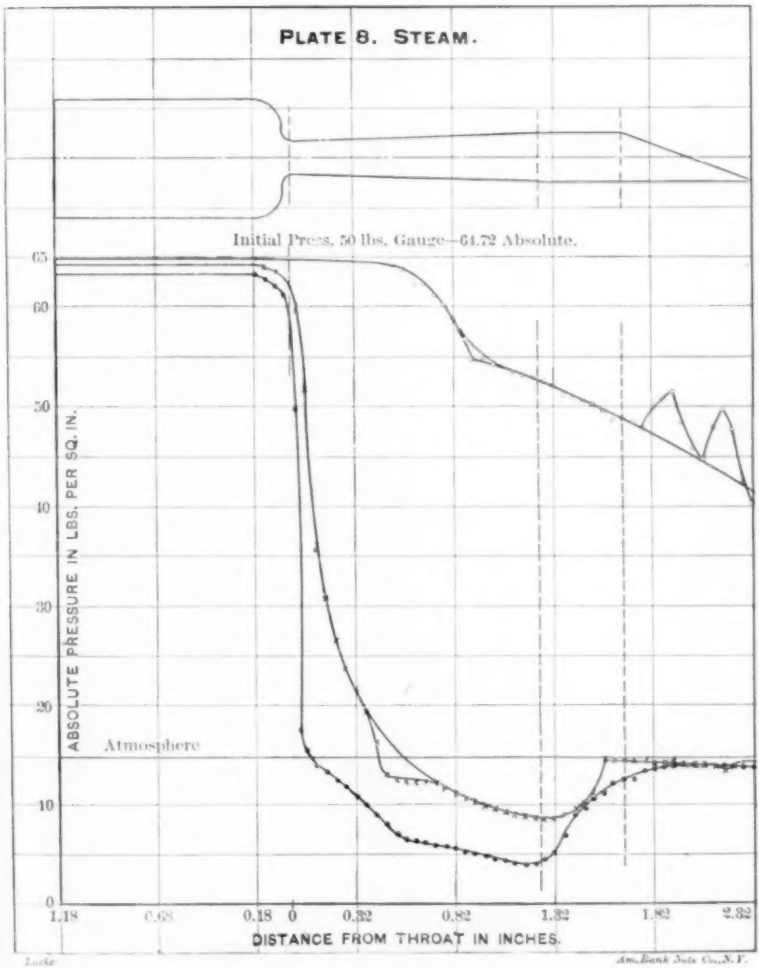
PLATE 8. STEAM.
Initial Press. 50 lbs. Gauge—64.72 Absolute.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
65
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 15.

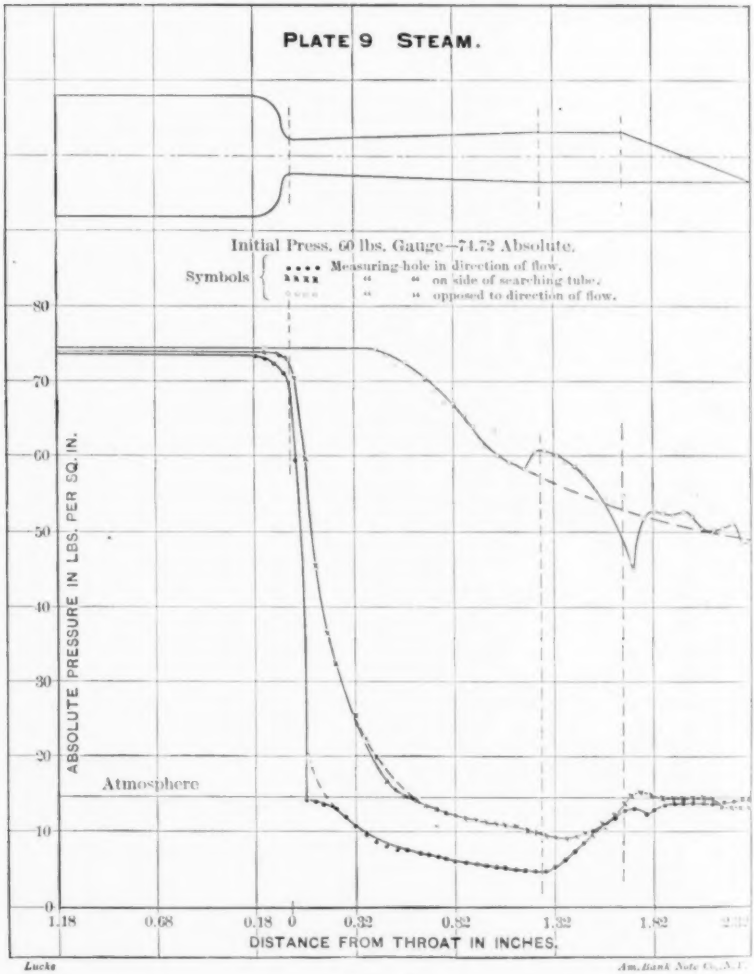
PLATE 9 STEAM.
Initial Press. 60 lbs. Gauge—74.72 Absolute.
Symbols
Measuring-hole in direction of flow.
" " on side of searching tube.
" " opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
80
70
60
50
40
30
20
Atmosphere
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 16.

AIR—INITIAL PRESSURE (70-POUND GAUGE — 84.72 ABSOLUTE).

DISTANCE FROM THROAT IN INCHES.	MEASURING HOLE IN THE DIRECTION OF FLOW.		MEASURING HOLE ON SIDE OF SEARCHING TUBE.		MEASURING HOLE OPPOSED TO DIRECTION OF FLOW.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	82.22	.....	83.72	.....	84.22
−0.43	.....	82.22	.....	83.72	.....	84.22
−0.38	.....	82.22	.....	83.72	.....	84.22
−0.33	.....	82.22	.....	83.72	.....	84.22
−0.28	.....	82.22	.....	83.72	.....	84.22
−0.23	.....	82.22	.....	83.72	.....	84.22
−0.18	.....	82.22	.....	83.72	.....	84.22
−0.13	.....	82.22	.....	83.72	.....	84.22
−0.08	.....	82.22	.....	83.72	.....	84.22
−0.03	.....	81.72	.....	83.72	.....	84.22
+0.02	.....	78.72	.....	81.72	.....	84.22
0.07	34.05	16.78	.....	76.22	.....	84.22
0.12	30.05	14.72	.....	57.72	.....	84.22
0.17	25.05	12.26	.....	42.72	.....	84.22
0.22	23.75	11.63	.....	37.22	.....	84.22
0.27	22.65	11.09	.....	30.22	.....	84.22
0.32	19.65	9.61	.....	26.22	.....	84.22
0.37	17.15	8.39	.....	22.22	.....	84.22
0.42	15.25	7.45	.....	17.08	.....	84.22
0.47	14.35	7.01	34.85	16.29	.....	83.72
0.52	13.85	6.77	33.25	15.75	.....	83.22
0.57	13.25	6.47	32.15	14.52	.....	82.72
0.62	12.75	6.23	29.65	13.74	.....	81.72
0.67	12.05	5.88	28.05	13.15	.....	80.72
0.72	11.45	5.59	26.85	12.31	.....	79.72
0.77	10.75	5.24	25.15	11.63	.....	78.72
0.82	10.25	5.00	23.65	11.18	.....	77.72
0.87	9.85	4.80	22.85	10.60	.....	76.72
0.92	9.75	4.75	21.65	10.30	.....	75.72
0.97	9.25	4.51	21.05	10.20	.....	74.02
1.02	8.95	4.36	20.85	9.71	.....	72.02
1.07	8.35	4.07	19.85	9.42	.....	71.52
1.12	8.05	3.92	19.25	9.17	.....	71.02
1.17	7.75	3.97	18.75	8.88	.....	70.02
1.22	8.75	3.96	18.15	8.58	.....	70.52
1.27	9.25	4.51	17.55	8.34	.....	70.52
1.32	9.65	4.70	14.85	8.24	.....	70.52
1.37	16.85	8.24	14.25	7.94	.....	70.52
1.42	18.65	9.13	17.25	8.44	.....	70.02
1.47	20.25	9.92	19.25	9.42	.....	70.02
1.52	21.35	10.45	20.85	10.20	.....	69.42
1.57	22.35	10.94	23.65	11.58	.....	69.42
1.62	23.45	11.48	28.05	13.74	.....	67.22
1.67	25.75	12.61	31.05	15.21	.....	66.72
1.72	25.15	12.31	29.55	14.47	.....	63.72
1.77	24.55	12.02	32.45	15.90	.....	56.72
1.82	24.55	12.02	31.85	15.60	.....	67.72
1.87	24.25	11.87	26.85	13.15	.....	73.72
1.92	22.05	10.79	29.15	14.28	.....	71.72
1.97	23.05	11.28	31.45	15.41	.....	66.72
2.02	23.45	11.48	29.25	14.33	.....	57.72
2.07	24.15	11.82	27.25	13.35	.....	51.72
2.12	25.25	12.36	28.05	13.74	.....	57.22
2.17	26.65	13.05	29.45	14.43	.....	65.72
2.22	29.35	14.38	28.05	13.74	.....	64.72
2.27	29.55	14.47	27.65	13.54	.....	61.22
2.32	29.45	14.43	27.75	13.59	.....	61.72

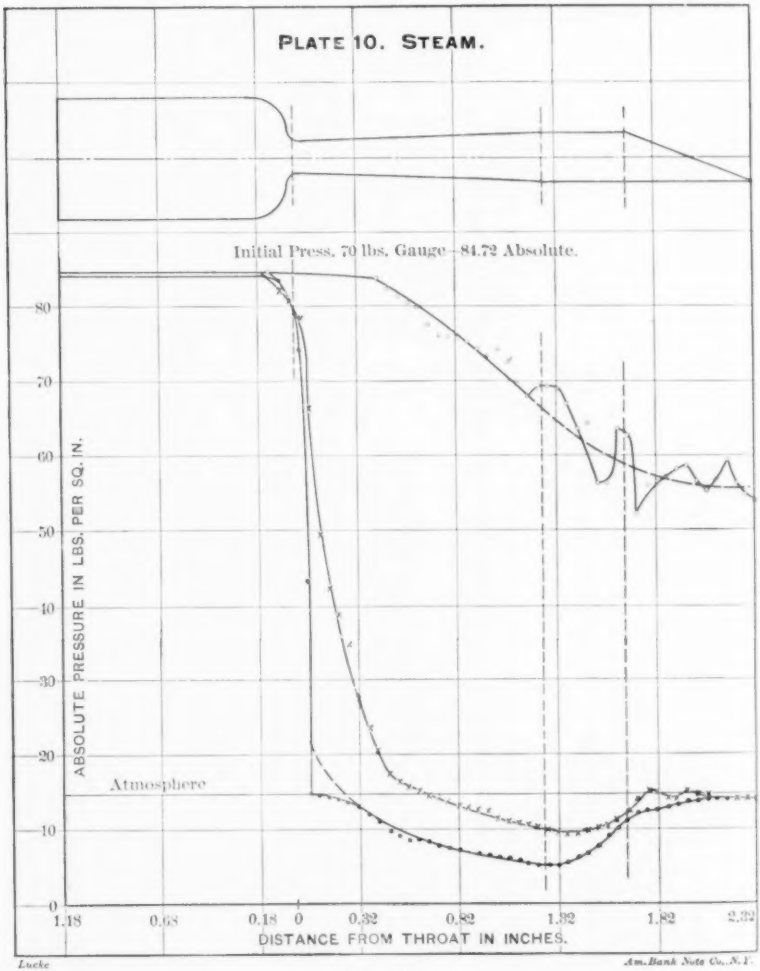
PLATE 10. STEAM.
Initial Press. 70 lbs. Gauge = 84.72 Absolute.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
80
70
60
50
40
30
20
Atmosphere
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 17.

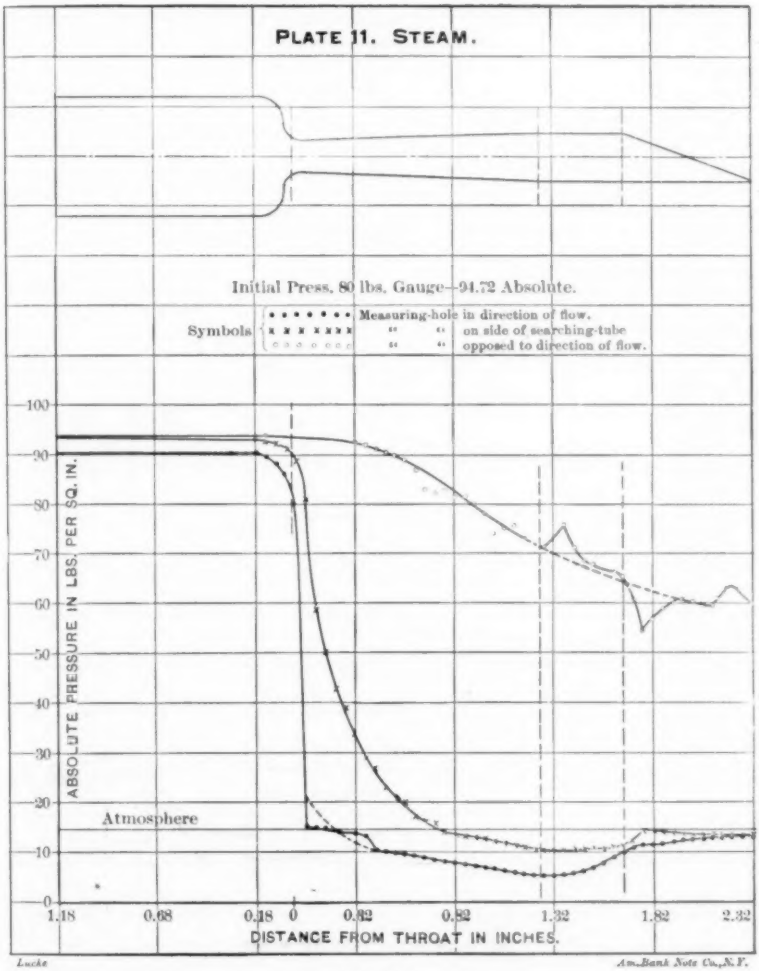
PLATE 11. STEAM.
Initial Press. 80 lbs. Gauge—94.72 Absolute.
Symbols
Measuring-hole in direction of flow.
" " on side of searching-tube
" " opposed to direction of flow.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.82
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 18.

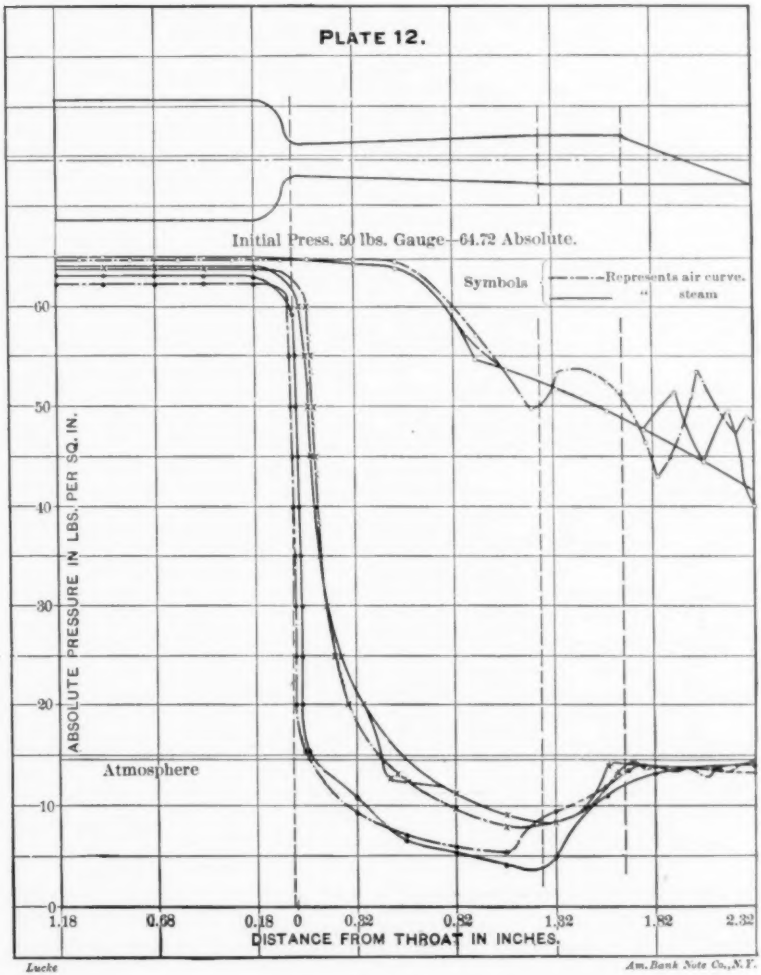
PLATE 12.
Initial Press. 50 lbs. Gauge—64.72 Absolute.
Symbols
Represents air curve.
" steam
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 19.

AIR—INITIAL PRESSURE (80-POUND GAUGE — 94.72 ABSOLUTE).

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
0.48	.....	91.22	.....	93.22	.....	94.22
−0.43	.....	91.22	.....	93.22	.....	94.22
−0.38	.....	91.22	.....	93.22	.....	94.22
−0.33	.....	91.22	.....	93.22	.....	94.22
−0.28	.....	91.22	.....	93.22	.....	94.22
−0.23	.....	91.22	.....	93.22	.....	94.22
−0.18	.....	91.22	.....	93.22	.....	94.22
−0.13	.....	91.22	.....	93.22	.....	94.22
−0.08	.....	91.22	.....	93.22	.....	94.22
−0.03	.....	90.22	.....	93.22	.....	94.22
+0.02	.....	87.72	.....	90.22	.....	94.22
0.07	36.05	17.67	.....	79.72	.....	94.22
0.12	35.05	17.18	.....	69.72	.....	94.22
0.17	28.45	13.93	.....	47.72	.....	94.22
0.22	25.75	12.61	.....	38.22	.....	94.22
0.27	24.55	12.02	.....	31.72	.....	94.22
0.32	21.25	10.40	.....	27.22	.....	94.22
0.37	19.05	9.32	.....	23.22	.....	94.22
0.42	16.85	8.24	.....	20.72	.....	94.22
0.47	15.65	7.67	35.05	17.18	.....	93.22
0.52	14.85	7.26	33.75	16.54	.....	92.72
0.57	14.45	7.06	32.05	15.70	.....	90.72
0.62	13.95	6.81	29.75	14.57	.....	89.22
0.67	13.25	6.47	28.45	13.93	.....	89.22
0.72	12.55	6.12	27.15	13.30	.....	87.22
0.77	11.75	5.73	25.25	12.36	.....	85.22
0.82	11.15	5.44	24.25	11.87	.....	84.22
0.87	10.65	5.19	22.95	11.23	.....	82.72
0.92	10.35	5.05	22.15	10.84	.....	81.72
0.97	10.15	4.95	21.75	10.64	.....	79.22
1.02	9.65	4.70	20.85	10.20	.....	77.72
1.07	9.25	4.51	20.15	9.86	.....	78.72
1.12	8.65	4.21	19.55	9.56	.....	80.22
1.17	8.25	4.02	18.75	9.17	.....	80.02
1.22	8.05	3.92	18.25	8.93	.....	79.72
1.27	8.45	4.11	17.65	8.63	.....	79.72
1.32	8.55	4.16	17.35	8.48	.....	78.72
1.37	9.75	4.75	16.65	8.14	.....	77.72
1.42	14.05	6.86	16.45	8.04	.....	74.72
1.47	18.15	8.88	17.95	8.78	.....	71.72
1.52	19.85	9.71	19.75	9.66	.....	70.22
1.57	21.35	10.45	21.25	10.40	.....	69.22
1.62	21.85	10.69	26.95	13.20	.....	64.72
1.67	22.95	11.25	29.85	14.62	.....	68.72
1.72	25.85	12.66	32.45	15.90	.....	60.22
1.77	25.45	12.46	29.25	14.33	.....	67.22
1.82	23.65	11.58	32.05	15.70	.....	63.72
1.87	23.15	11.33	34.65	16.98	.....	63.72
1.92	21.25	10.40	28.65	14.03	.....	59.72
1.97	20.45	10.01	27.05	13.25	.....	54.72
2.02	20.85	10.20	28.85	14.13	.....	58.72
2.07	21.65	10.60	30.65	15.01	.....	61.72
2.12	22.05	10.79	29.75	14.57	.....	57.72
2.17	22.75	11.14	28.15	13.79	.....	57.72
2.22	27.95	13.69	28.25	13.84	.....	61.72
2.27	29.35	14.38	28.45	13.93	.....	59.72
2.32	29.35	14.38	27.55	13.49	.....	57.72

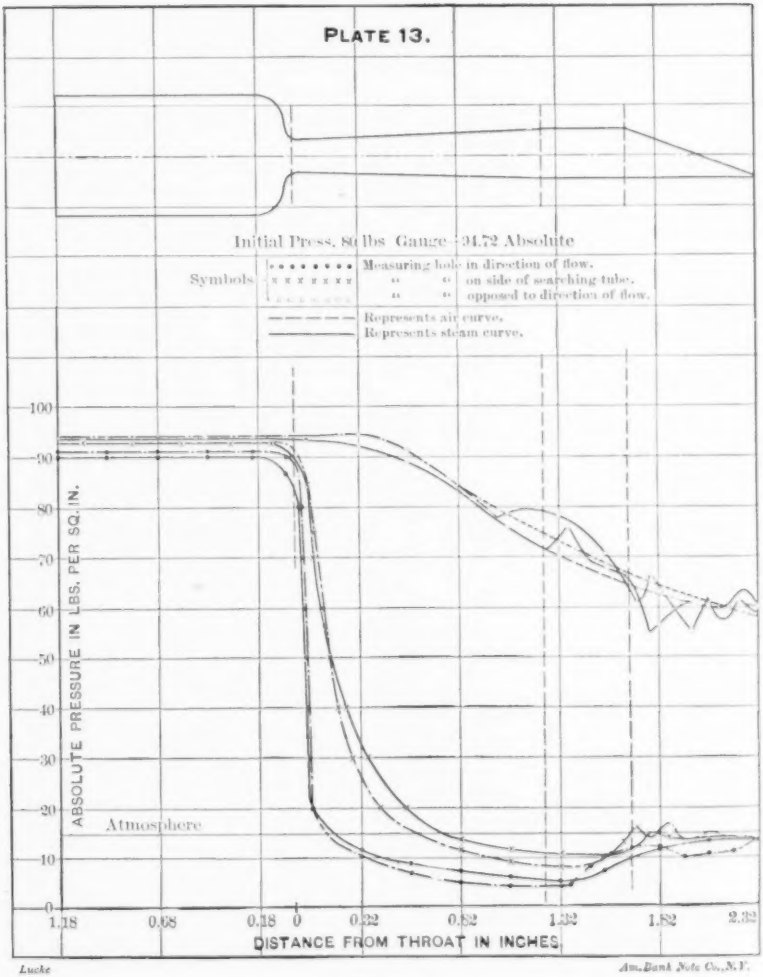
PLATE 13.
Initial Press. 80 lbs Gauge = 94.72 Absolute
Symbols
Measuring hole in direction of flow.
" " on side of searching tube.
" " opposed to direction of flow.
Represents air curve.
Represents steam curve.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 20.

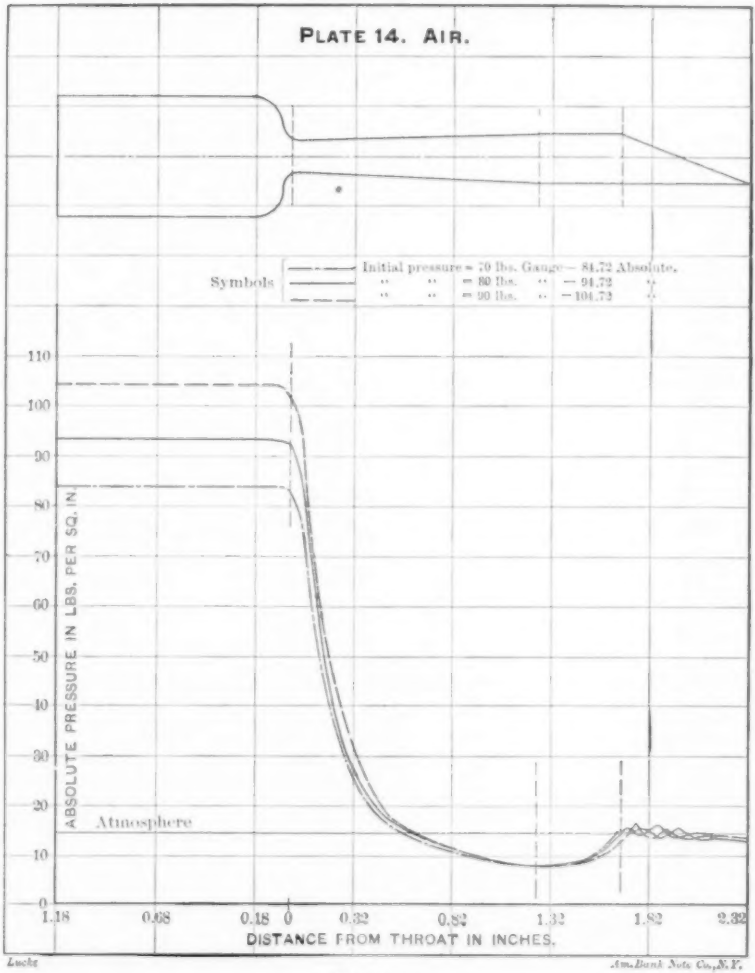
PLATE 14. AIR.
Symbols
Initial pressure = 70 lbs. Gauge — 84.72 Absolute.
" " = 80 lbs. " — 94.72 "
" " = 90 lbs. " — 104.72 "
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
110
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Luehr
Am. Bank Note Co., N.Y.

FIG. 21.

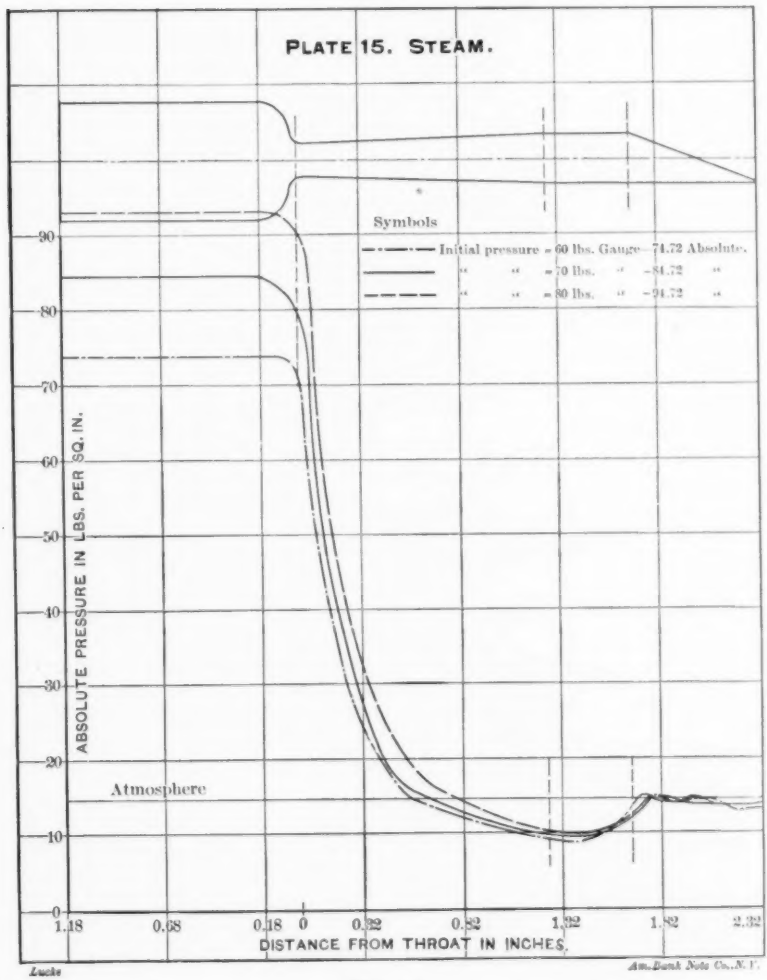
PLATE 15. STEAM.
Symbols
Initial pressure = 60 lbs. Gauge = 74.72 Absolute.
" " = 70 lbs. " = 84.72 "
" " = 80 lbs. " = 94.72 "
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
90
80
70
60
50
40
30
20
Atmosphere
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 22.

AIR—INITIAL PRESSURE (90-POUND GAUGE — 104.72 ABSOLUTE).

DISTANCE FROM THROAT IN INCHES.	MEASURING HOLE IN THE DIRECTION OF FLOW.		MEASURING HOLE ON SIDE OF SEARCHING TUBE.		MEASURING HOLE OPPOSED TO DIRECTION OF FLOW.	
	Absolute Pressure in Ins. of hgt.	Absolute Pressure in lbs. per sq. in	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	101 22	.....	104.72	.....	104.72
−0.43	.....	101.22	.....	104.72	.....	104.72
−0.38	.....	101.22	.....	104.72	.....	104.72
−0.33	.....	101.22	.....	104.72	.....	104.72
−0.28	.....	101.22	.....	104.72	.....	104.72
−0.23	.....	101.22	.....	104.72	.....	104.72
−0.18	.....	101.22	.....	104.72	.....	104.72
−0.13	.....	101.22	.....	104.72	.....	104.72
−0.08	.....	101.22	.....	104.72	.....	104.72
−0.03	.....	100.72	.....	104.72	.....	104.72
+0.02	.....	97.72	.....	99.72	.....	103.72
0.07	39.05	19.14	.....	94.72	.....	103.72
0.12	33.55	16.44	.....	68.22	.....	103.72
0.17	30.05	14.72	.....	55.72	.....	103.72
0.22	29.25	14.33	.....	45.22	.....	103.72
0.27	28.55	13.98	.....	38 22	.....	103.72
0.32	26.05	12.76	.....	33.22	.....	103.72
0.37	23.05	11.28	.....	28.22	.....	103.72
0.42	20 45	10.01	.....	24.22	.....	103.72
0.47	18.45	9.02	37.35	18.30	.....	102.72
0.52	15.85	7.75	35.85	17.57	.....	102.22
0.57	15.20	7.43	34.25	16.78	.....	100.72
0.62	14.85	7.26	31.75	15.55	.....	98.72
0.67	14.05	6.86	31.35	14.87	.....	95.72
0.72	13.50	6.59	29.05	14.23	.....	92.22
0.77	12.85	6.27	27 15	13.30	.....	90.72
0.82	12.10	5.91	25.65	12.56	.....	90.22
0.87	11.60	5.66	24.15	11.82	.....	88.72
0.92	11.15	5.44	23.25	11.38	.....	86.72
0.97	10.85	5.30	23.05	11.28	.....	85.22
1.02	10.50	5 12	22.05	10.79	.....	88.72
1.07	10.05	4.84	21.25	10 40	.....	90.22
1 12	9.55	4.65	20.55	10.06	.....	89.72
1.17	9.25	4.51	19.75	9.66	.....	89.22
1.22	7 55	3.65	19.05	9.32	.....	85.72
1.27	3.05	1.46	18.65	9.12	.....	84.72
1.32	8.05	3.92	18.35	8.98	.....	78.72
1.37	11.15	5.44	17.65	8.63	.....	75.22
1.42	11.85	5.78	17.05	8.34	.....	74.72
1.47	16.05	7.85	17.75	8.68	.....	63.72
1.52	19.65	9.61	19.35	9.47	.....	69.72
1.57	20.65	10.10	20.05	9.81	.....	70.72
1.62	21.15	10.35	22.25	10.89	.....	68.72
1.67	22.35	10.94	26.05	12.76	.....	66.72
1 72	24 35	11.92	31.45	15.41	.....	53.22
1.77	27.05	13.25	35.25	17.27	.....	70.22
1.82	24.55	12.02	29.85	14.62	.....	85.22
1.87	21.15	10.35	29.55	14.47	.....	83.72
1.92	20.45	10.01	32.85	16.09	.....	83.72
1.97	21.50	10.52	30.35	14.87	.....	88.72
2.02	22.00	10.77	29.15	14 28	.....	88.22
2.07	23.00	11.26	28.25	13.84	.....	84.72
2.12	23.20	11.36	28.85	14.13	.....	82.72
2.17	28.35	13.89	29.25	14.33	.....	79.22
2.22	29.55	14.47	29.65	14.52	.....	75.72
2.27	29.45	14.43	28 55	13.98	.....	66.72
2.32	24.35	14.38	27.75	13.59	.....	65.72

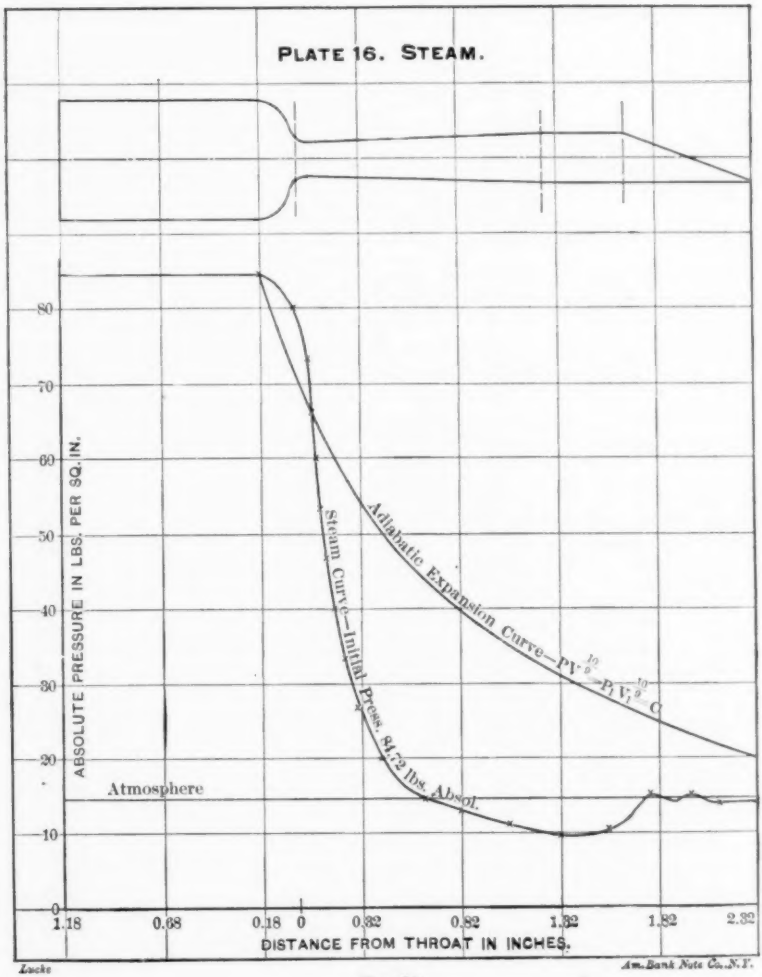
PLATE 16. STEAM.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
80
70
60
50
40
30
20
10
0
Atmosphere
Adiabatic Expansion Curve—$PV^{\frac{10}{9}}=P_1V_1^{\frac{10}{9}}C$
Steam Curve—Initial Press. 84.72 lbs. Absol.
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co. N.Y.

FIG. 23.

PLATE 17. STEAM.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
Steam Curve—Initial Press. 94.72 Absol.
Adiabatic Expansion Curve—$PV^{\frac{10}{9}}=P_1V_1^{\frac{10}{9}}=C$
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N.Y.

FIG. 24.

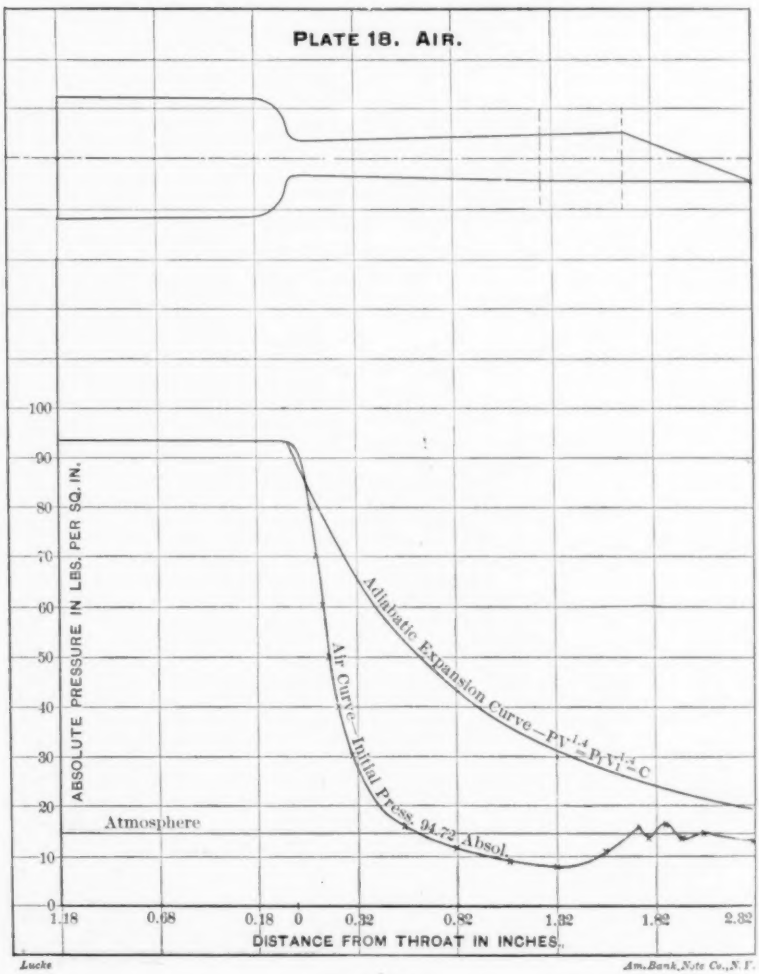
PLATE 18. AIR.
ABSOLUTE PRESSURE IN LBS. PER SQ. IN.
100
90
80
70
60
50
40
30
20
10
0
Atmosphere
Adiabatic Expansion Curve—$PV^{1.4}=P_1V_1^{1.4}=C$
Air Curve—Initial Press. 94.73 Absol.
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 25.

STEAM—INITIAL PRESSURE (40-POUND GAUGE — 54.72 ABSOLUTE).

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	53.22	.....	54.47	.....	54.72
−0.43	.....	53.22	.....	54.47	.....	54.72
−0.38	.....	53.22	.....	54.47	.....	54.72
−0.33	.....	53.22	.....	54.47	.....	54.72
−0.28	.....	53.22	.....	54.47	.....	54.72
−0.23	.....	53.22	.....	54.47	.....	54.72
−0.18	.....	53.22	.....	54.57	.....	54.72
−0.13	.....	53.22	.....	54.47	.....	54.72
−0.08	.....	52.72	.....	53.72	.....	54.72
−0.03	.....	52.72	.....	53.22	.....	54.72
+0.02	.....	46.72	.....	51.72	.....	54.72
0.07	29.35	14.38	.....	44.72	.....	54.72
0.12	27.25	13.35	.....	32.22	.....	54.47
0.17	25.85	12.66	.....	26.72	.....	54.47
0.22	22.85	11.18	.....	23.72	.....	54.47
0.27	21.35	10.45	.....	21.72	.....	54.47
0.32	19.75	9.67	.....	20.72	.....	54.47
0.37	17.45	8.53	30.05	14.72	.....	54.47
0.42	15.85	7.75	29.45	14.43	.....	54.47
0.47	14.25	6.96	28.65	14.03	.....	54.22
0.52	13.45	6.57	27.35	13.39	.....	53.47
0.57	13.05	6.37	25.95	12.71	.....	53.47
0.62	12.55	6.13	24.45	11.97	.....	53.22
0.67	12.05	5.88	23.55	11.53	.....	52.22
0.72	11.65	5.69	22.45	10.99	.....	50.22
0.77	11.05	5.39	21.55	10.55	.....	49.72
0.82	10.75	5.24	20.55	10.06	.....	48.22
0.87	10.45	5.10	19.85	9.71	.....	46.97
0.92	9.95	4.85	19.35	9.47	.....	46.22
0.97	9.55	4.65	18.75	9.17	.....	45.22
1.02	9.35	4.56	18.25	8.93	.....	44.72
1 07	8.95	4.36	17.85	8.73	.....	45.72
1.12	9 35	4.56	17.25	8.44	.....	44.22
1.17	10.15	4.95	16.85	8.24	.....	43.72
1.22	11.25	5.49	16.75	8.19	.....	43.72
1.27	16.45	8.04	17.95	8.78	.....	43.22
1.32	18.45	9.02	19.05	9.32	.....	42.72
1.37	19.85	9.72	20.35	9.96	.....	42.47
1.42	21.55	10.55	21.45	10.50	.....	42.47
1.47	22 45	10.99	24.25	11.87	.....	42.22
1.52	23.95	11.72	26.05	12.76	.....	41.72
1.57	25.85	12.66	28.55	13.98	.....	40.22
1.62	27.85	14.64	29.15	14.28	.....	41.22
1.67	28.35	13.89	28.55	13.98	.....	41.22
1.72	28.85	14.13	29.05	14.23	.....	43.72
1.77	28 95	14 18	28.55	13.98	.....	43.22
1.82	28.45	13.93	28.45	13.93	.....	40.72
1.87	29.75	14.57	28.35	13.89	.....	37.72
1.92	29.75	14.57	28.15	13.79	.....	37.72
1.97	29.85	14.62	28 15	13.79	.....	41.22
2.02	29.65	14.52	28.15	13.79	....	40.72
2.07	29.65	14.52	28.25	13.84	.....	40.72
2.12	29.65	14.52	28.35	13.89	.....	37.72
2.17	29.65	14.52	28.45	13.93	.....	36.72
2.22	29.65	14.52	28.65	14.03	.....	36.22
2.27	29.65	14.52	28.75	14.08	.....	34.72
2.32	29.45	14.43	28.75	14.08	.....	34.72

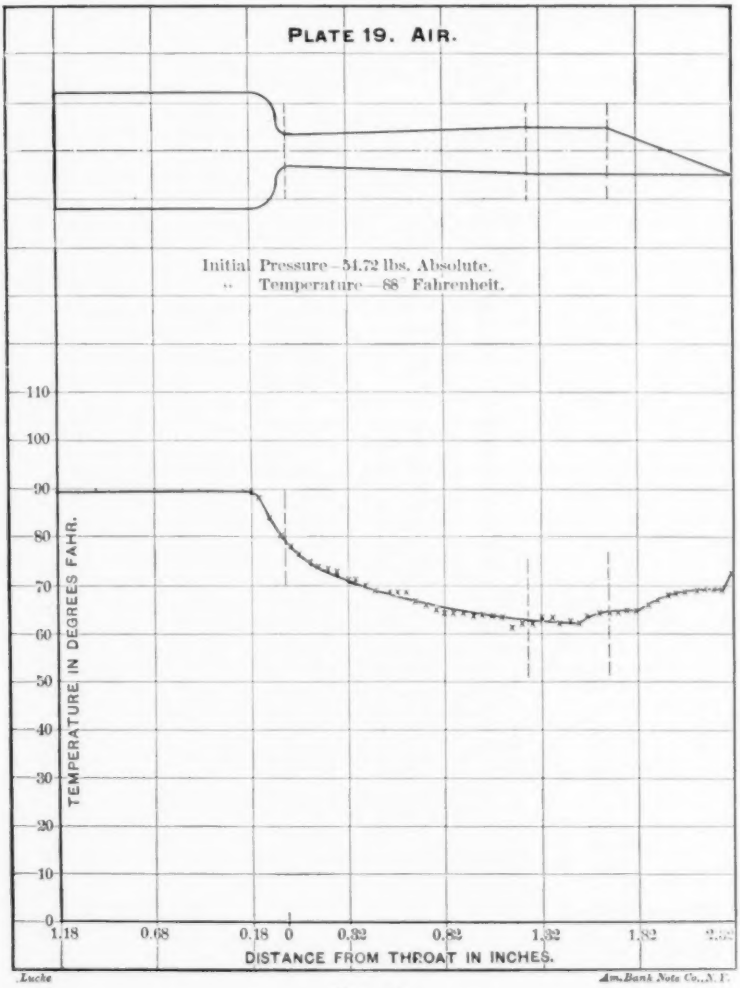
PLATE 19. AIR.
Initial Pressure—54.72 lbs. Absolute.
" Temperature—88° Fahrenheit.
TEMPERATURE IN DEGREES FAHR.
110
100
90
80
70
60
50
40
30
20
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N. Y.

FIG. 26.

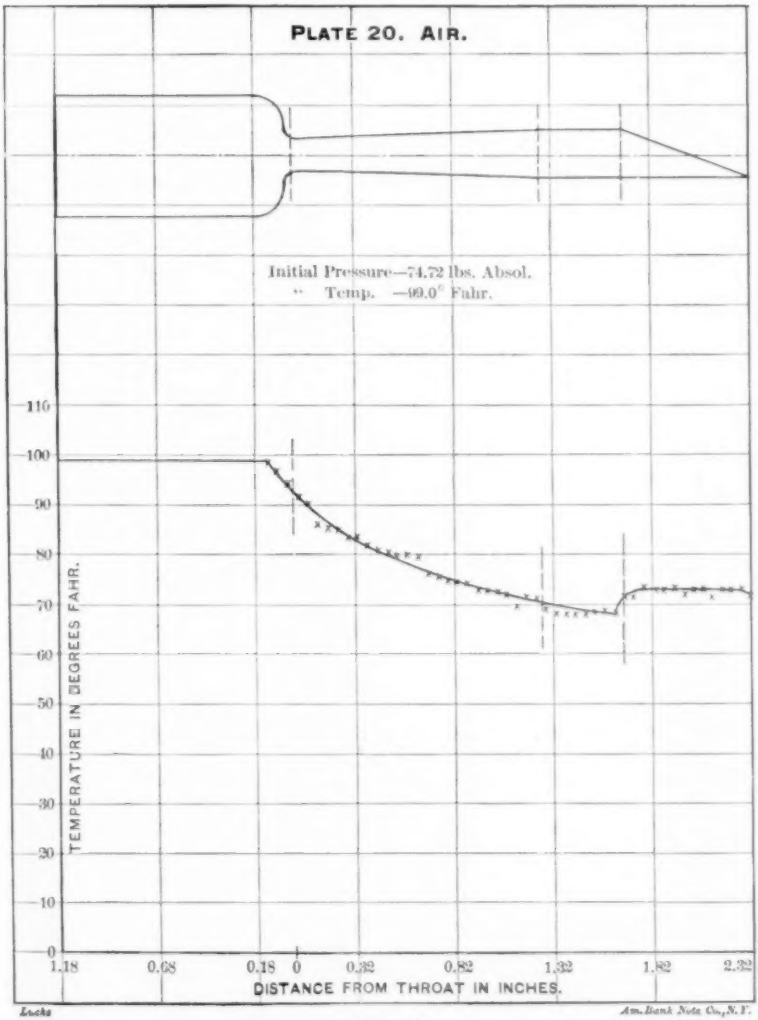
PLATE 20. AIR.
Initial Pressure—74.72 lbs. Absol.
" Temp. —99.0° Fahr.
TEMPERATURE IN DEGREES FAHR.
110
100
90
80
70
60
50
40
30
20
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Locke
Am. Bank Note Co., N.Y.

FIG. 27.

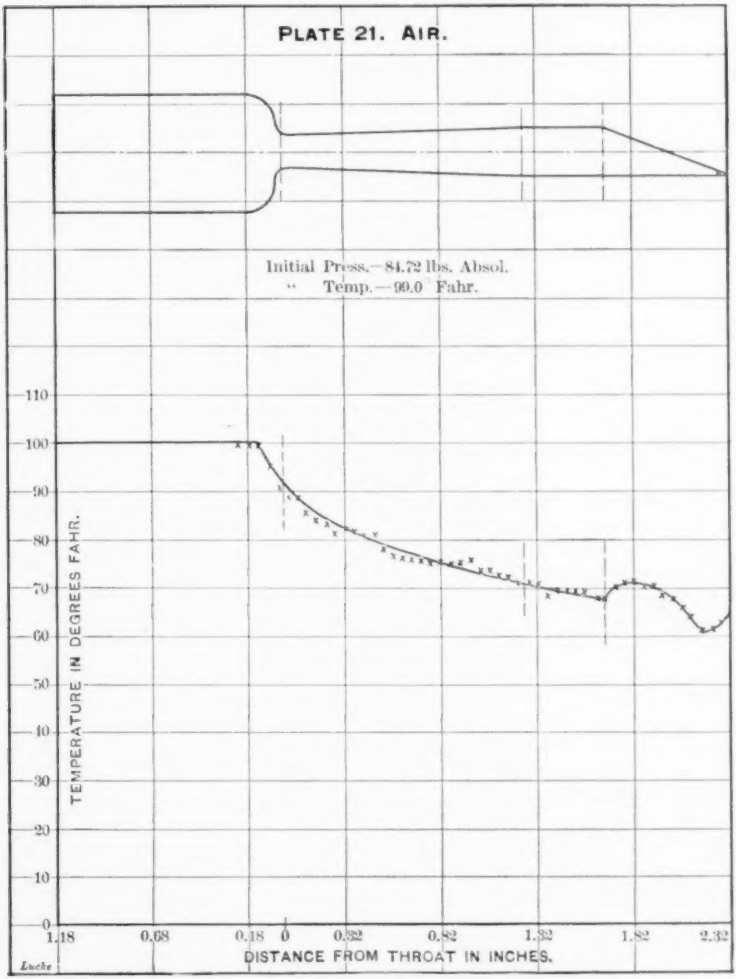
PLATE 21. AIR.
Initial Press.—84.72 lbs. Absol.
" Temp.—99.0 Fahr.
TEMPERATURE IN DEGREES FAHR.
110
100
90
80
70
60
50
40
30
20
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke

FIG. 28.

STEAM—INITIAL PRESSURE (50-POUND GAUGE — 64.72 ABSOLUTE).

DISTANCE FROM THROAT IN INCHES.	MEASURING HOLE IN THE DIRECTION OF FLOW.		MEASURING HOLE ON SIDE OF SEARCHING TUBE.		MEASURING HOLE OPPOSED TO DIRECTION OF FLOW.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	63.22	.....	64.22	.....	64.72
−0.43	.....	63.22	.....	64.22	.....	64.72
−0.38	.....	63.22	.....	64.22	.....	64.72
−0.33	.....	63.22	.....	64.22	.....	64.72
−0.28	.....	68.22	.....	64.22	.....	64.72
−0.23	.....	63.22	.....	64.22	.....	64.72
−0.18	.....	63.22	.....	64.22	.....	64.72
−0.13	.....	62.72	.....	64.22	.....	64.22
−0.08	.....	62 22	.....	63.72	.....	64.22
−0.03	.....	61.22	.....	62.72	.....	64 22
+0.02	.....	49.72	.....	59.72	.....	64.22
0.07	31.45	15.41	.....	51.72	.....	64.22
0.12	28.25	13.84	.....	35.72	.....	64.22
0.17	27.65	13.54	.....	30.72	.....	64.22
0.22	25.85	12.66	.....	26.72	.....	64.22
0.27	24.65	12.07	.....	23.72	.....	64.22
0.32	22.15	10.84	.....	21.72	.....	64.42
0.37	20.65	10.10	.....	19.22	.....	64.22
0.42	18.25	8.93	33.85	16.59	.....	64.22
0.47	16.45	8.04	26.45	12.95	.....	64.22
0.52	14.75	7.21	26.25	12.85	.....	64.22
0.57	13.65	6.67	26 15	12.81	.....	63.22
0.62	13.25	6.47	26.05	12.76	.....	62.22
0.67	12.95	6.32	25.65	12.56	.....	62.22
0.72	12.15	5 93	25.25	12.36	.....	61.22
0.77	11.75	5.72	24.05	11.77	.....	60.22
0.82	11.05	5.39	23.05	11.28	.....	58.72
0.87	10.75	5.24	22.05	10.79	.....	56.72
0.92	10.35	5.05	21.45	10.50	.....	54.72
0.97	9.85	4.80	20 75	10.15	.....	54.72
1.02	9.45	4.61	20.05	9.81	.....	54.47
1.07	8.85	4.31	19.65	9.61	.....	54.22
1.12	8.45	4.11	19.05	9.32	.....	53.47
1.17	8.15	4.97	18.45	9.02	.....	53.22
1.22	8.35	4.07	18.05	8 83	.....	53.22
1.27	9.05	4.41	17.65	8.63	.....	52.72
1.32	10.85	5.29	17.35	8.48	.....	52.22
1.37	14.45	7.06	18.35	8.98	.....	51.22
1.42	18.45	9.02	19.85	9.71	.....	50.72
1.47	20.05	9.81	21.05	10.30	.....	50.47
1.52	21.55	10.55	22.25	10.89	.....	50.22
1.57	22.65	11.09	29.85	14.62	.....	49.72
1.62	25.15	12.31	29 85	14.62	.....	48.72
1.67	25.25	12.36	29.75	14.57	.....	48.72
1.72	25.85	12.66	29 65	14.52	.....	47.22
1.77	27.35	13.39	29.25	14.33	.....	47.72
1.82	28.05	13.74	28.55	13.98	.....	50.22
1.87	28.45	14.93	29.25	14.33	.....	50.73
1.92	28.65	14.03	29.25	14.33	.....	51.47
1.97	28.65	14.03	28.55	13.98	.....	48.22
2.02	28.65	14.03	28.65	14.03	.....	45.72
2.07	28.55	13.98	28.65	14.03	.....	44.72
2.12	28.55	13.98	28.45	13.93	.....	47.72
2.17	28.55	13.98	28.35	13.89	.....	44.72
2.22	28.55	13.98	28.45	13.93	.....	36 22
2.27	28.55	13.98	28.65	14.03	.....	42.72
2.32	28.25	13.84	28.75	14.08	.....	40.22

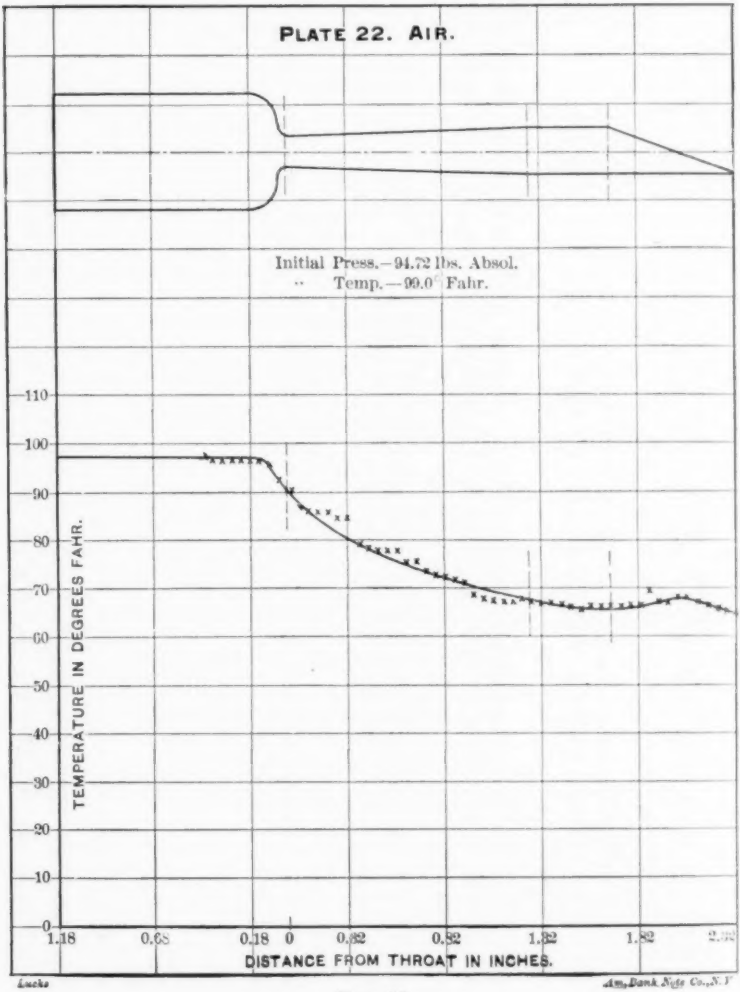
PLATE 22. AIR.
Initial Press.—94.72 lbs. Absol.
" Temp.—99.0° Fahr.
TEMPERATURE IN DEGREES FAHR.
110
100
90
80
70
60
50
40
30
20
10
0
1.18
0.68
0.18
0
0.82
0.82
1.32
1.82
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N. Y.

FIG. 29.

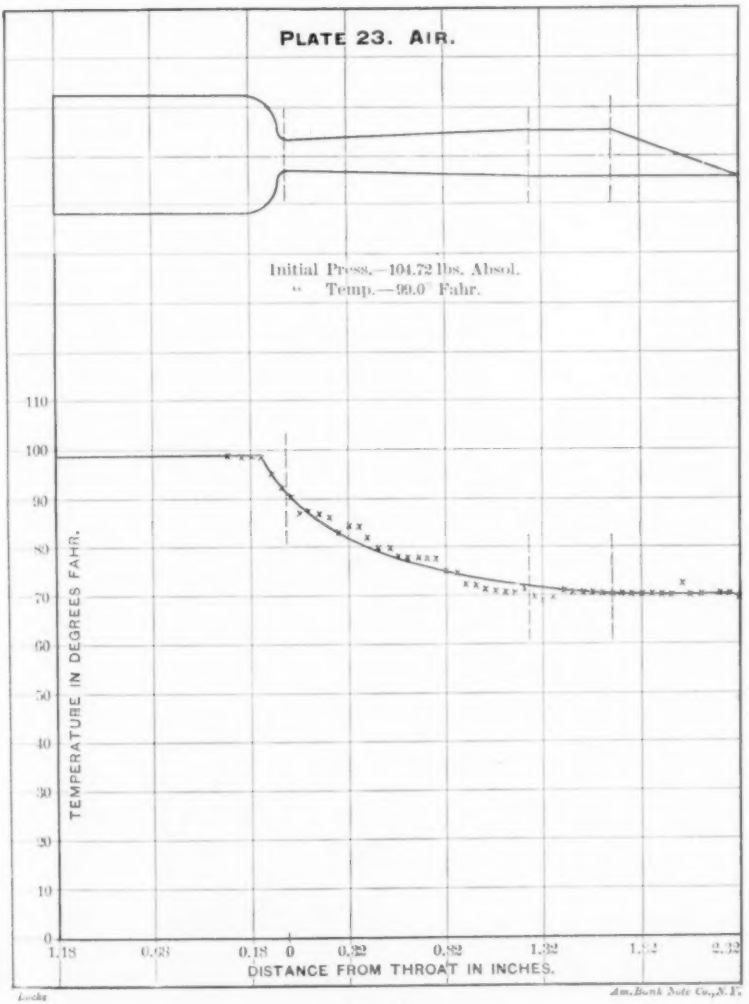
PLATE 23. AIR.
Initial Press.—104.72 lbs. Absol.
" Temp.—99.0° Fahr.
TEMPERATURE IN DEGREES FAHR.
110
100
90
80
70
60
50
40
30
20
10
0
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N. Y.

FIG. 30.

STEAM—INITIAL PRESSURE (60-POUND GAUGE — 74.72 ABSOLUTE).

DISTANCE FROM THROAT IN INCHES.	MEASURING HOLE IN THE DIRECTION OF FLOW.		MEASURING HOLE ON SIDE OF SEARCHING TUBE.		MEASURING HOLE OPPOSED TO DIRECTION OF FLOW.	
	Absolute Pressure in Ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in Ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in Ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	73.72	.....	73.97	.....	74.72
−0.43	.....	73.72	.....	73.97	.....	74.72
−0.38	.....	73.72	.....	73.97	.....	74.72
−0.33	.....	73.72	.....	73.97	.....	74.72
−0.28	.....	73.72	.....	73.97	.....	74.72
−0.23	.....	73.72	.....	73.97	.....	74.72
−0.18	.....	73.72	.....	73.97	.....	74.72
−0.13	.....	73.22	.....	73.97	.....	74.72
−0.08	.....	72.72	.....	73.72	.....	74.22
−0.03	.....	71.22	.....	73.22	.....	74.22
+0.02	.....	59.72	.....	70.22	.....	74.22
0.07	29.85	14.62	.....	59.72	.....	74.22
0.12	29.45	14.43	.....	45.72	.....	74.22
0.17	28.45	13.93	.....	36.72	.....	74.22
0.22	27.05	13.25	.....	32.72	.....	74.22
0.27	24.65	12.07	.....	29.72	.....	74.22
0.32	22.35	10.94	.....	25.72	.....	74.22
0.37	20.05	9.81	.....	22.72	.....	74.22
0.42	18.45	9.02	.....	20.22	.....	74.22
0.47	17.45	8.53	34.75	17.03	.....	73.72
0.52	16.55	8.09	32.25	15.80	.....	72.72
0.57	16.25	7.94	31.05	15.21	.....	72.22
0.62	15.65	7.65	29.95	14.67	.....	71.22
0.67	15.15	7.40	29.05	14.23	.....	70.72
0.72	14.45	7.06	27.65	13.54	.....	68.72
0.77	14.05	6.86	26.45	12.95	.....	67.22
0.82	13.55	6.62	25.35	12.41	.....	67.22
0.87	12.95	6.32	24.35	11.92	.....	65.72
0.92	12.85	6.27	23.85	11.68	.....	64.22
0.97	12.65	6.18	22.85	11.18	.....	61.22
1.02	12.05	5.88	22.25	10.89	.....	63.72
1.07	11.55	5.64	21.65	10.60	.....	59.72
1.12	11.05	5.39	20.95	10.26	.....	59.22
1.17	10.65	5.19	20.45	10.01	.....	58.72
1.22	10.25	5.00	19.85	9.71	.....	61.22
1.27	10.05	4.90	19.35	9.47	.....	61.22
1.32	11.05	5.39	18.95	9.27	.....	60.22
1.37	13.05	6.37	18.65	9.12	.....	59.22
1.42	14.95	7.31	18.85	9.22	.....	58.72
1.47	18.25	8.93	19.95	9.76	.....	57.72
1.52	21.55	10.55	20.65	10.10	.....	55.22
1.57	22.85	11.18	22.65	11.09	.....	55.22
1.62	23.65	11.58	24.55	12.02	.....	54.22
1.67	26.45	12.95	29.45	14.43	.....	55.22
1.72	26.85	13.15	30.45	14.92	.....	46.22
1.77	25.85	12.66	30.25	14.82	.....	52.22
1.82	26.35	12.90	30.05	14.72	.....	52.97
1.87	28.85	14.13	29.65	14.52	.....	52.72
1.92	28.55	13.98	29.25	14.33	.....	52.22
1.97	28.35	13.89	29.25	14.33	.....	53.22
2.02	28.25	13.84	29.55	14.47	.....	52.22
2.07	28.45	13.93	29.25	14.33	.....	50.47
2.12	28.65	14.03	28.85	14.13	.....	50.22
2.17	28.95	14.18	28.65	14.03	.....	50.72
2.22	29.25	14.33	28.55	13.98	.....	51.22
2.27	29.45	14.43	28.35	13.89	.....	48.22
2.32	29.65	14.52	28.45	13.93	.....	40.72

PLATE 24. STEAM.
Initial Press.—54.72 lbs. Absol.
" Temp.—286.6 Fahr.
TEMPERATURE IN DEGREES FAHR.
310
320
330
340
350
360
370
380
390
400
410
420
430
440
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 31.

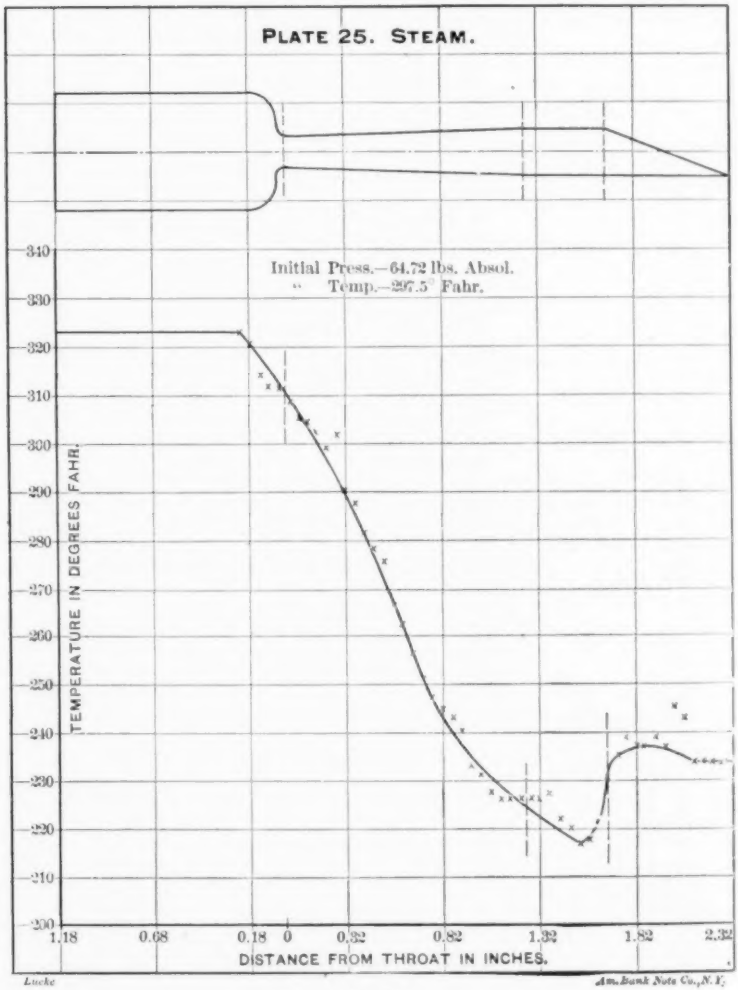
PLATE 25. STEAM.
Initial Press.—64.72 lbs. Absol.
" Temp.—297.5° Fahr.
TEMPERATURE IN DEGREES FAHR.
340
330
320
310
300
290
280
270
260
250
240
230
220
210
200
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Lucke
Am. Bank Note Co., N. Y.

FIG. 32.

STEAM—INITIAL PRESSURE (70-POUND GAUGE — 84.72 ABSOLUTE).

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	84.22	.....	84.22	.....	84.72
−0.43	.....	84.22	.....	84.22	.....	84.72
−0.38	.....	84.22	.....	84.22	.....	84.72
−0.33	.....	84.22	.....	84.22	.....	84.72
−0.28	.....	84.22	.....	84.22	.....	84.72
−0.23	.....	84.22	.....	84.22	.....	84.72
−0.18	.....	84.22	.....	84.22	.....	84.72
−0.13	.....	84.22	.....	84.22	.....	84.72
−0.08	.....	83.72	.....	82.22	.....	83.72
−0.03	.....	83.72	.....	81.72	.....	83.72
+0.02	.....	74.72	.....	78.72	.....	83.72
0.07	30.45	14.92	.....	66.32	.....	83.72
0.12	30.25	14.82	.....	49.72	.....	83.72
0.17	30.05	14.72	.....	42.22	.....	83.72
0.22	29.45	14.43	.....	38.22	.....	83.72
0.27	28.85	14.13	.....	34.72	.....	83.72
0.32	27.95	13.69	.....	27.22	.....	83.72
0.37	25.55	12.51	.....	23.72	.....	83.72
0.42	22.55	11.04	.....	20.72	.....	83.72
0.47	20.75	10.15	35.85	17.57	.....	82.22
0.52	19.15	9.37	33.65	16.49	.....	81.72
0.57	18.65	9.12	32.05	15.70	.....	80.72
0.62	18.05	8.83	31.25	15.31	.....	80.22
0.67	17.45	8.53	29.85	14.62	.....	77.72
0.72	16.75	8.19	29.25	14.33	.....	76.22
0.77	16.05	7.85	28.45	13.93	.....	76.22
0.82	15.45	7.55	27.25	13.35	.....	76.22
0.87	15.05	7.35	26.25	12.85	.....	75.72
0.92	14.25	6.96	25.65	12.56	.....	74.72
0.97	13.65	6.67	25.05	12.26	.....	73.72
1.02	13.35	6.52	24.05	11.77	.....	74.22
1.07	12.65	6.18	23.35	11.43	.....	72.72
1.12	12.15	5.93	22.55	11.04	.....	69.72
1.17	11.75	5.73	21.85	10.69	.....	68.22
1.22	11.35	5.54	21.25	10.40	.....	69.72
1.27	11.05	5.39	20.85	10.20	.....	69.22
1.32	11.05	5.39	20.35	9.96	.....	69.22
1.37	11.95	5.83	19.95	9.76	.....	66.72
1.42	13.25	6.47	20.05	9.82	.....	63.72
1.47	14.45	7.06	20.85	10.20	.....	64.72
1.52	16.25	7.94	21.45	10.50	.....	56.22
1.57	19.25	9.42	21.65	10.60	.....	56.72
1.62	21.45	10.50	22.25	10.89	.....	63.72
1.67	23.05	11.28	24.45	11.97	.....	62.72
1.72	25.45	12.46	29.25	14.33	.....	52.22
1.77	25.55	12.51	30.75	15.06	.....	56.22
1.82	25.15	12.31	30.25	14.82	.....	56.72
1.87	26.55	13.00	29.95	14.68	.....	57.72
1.92	28.25	13.84	29.75	14.58	.....	58.22
1.97	28.25	13.84	30.25	14.82	.....	59.72
2.02	28.85	14.13	29.85	14.62	.....	57.22
2.07	28.65	14.03	29.25	14.33	.....	56.22
2.12	28.85	14.13	29.15	14.28	.....	57.72
2.17	29.05	14.23	29.25	14.33	.....	59.72
2.22	29.25	14.33	29.05	14.23	.....	56.72
2.27	29.35	14.38	28.45	13.93	.....	54.47
2.32	29.35	14.38	28.45	13.93	.....	54.22

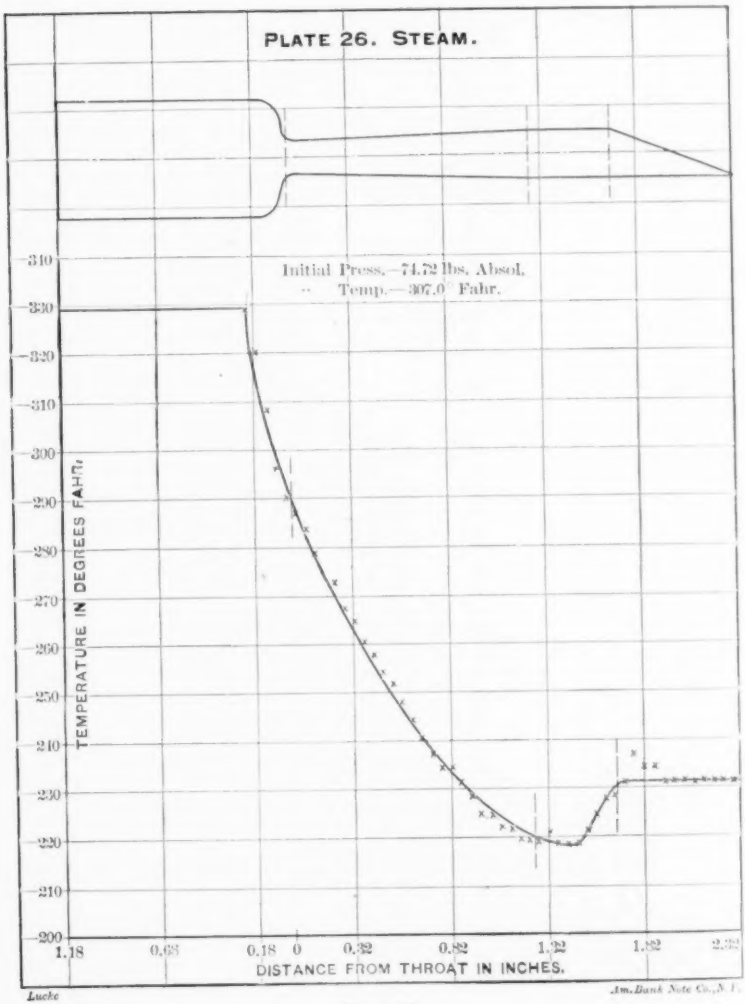
PLATE 26. STEAM.
Initial Press.—74.72 lbs. Absol.
" Temp.—307.0° Fahr.
TEMPERATURE IN DEGREES FAHR.
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N. Y.

FIG. 33.

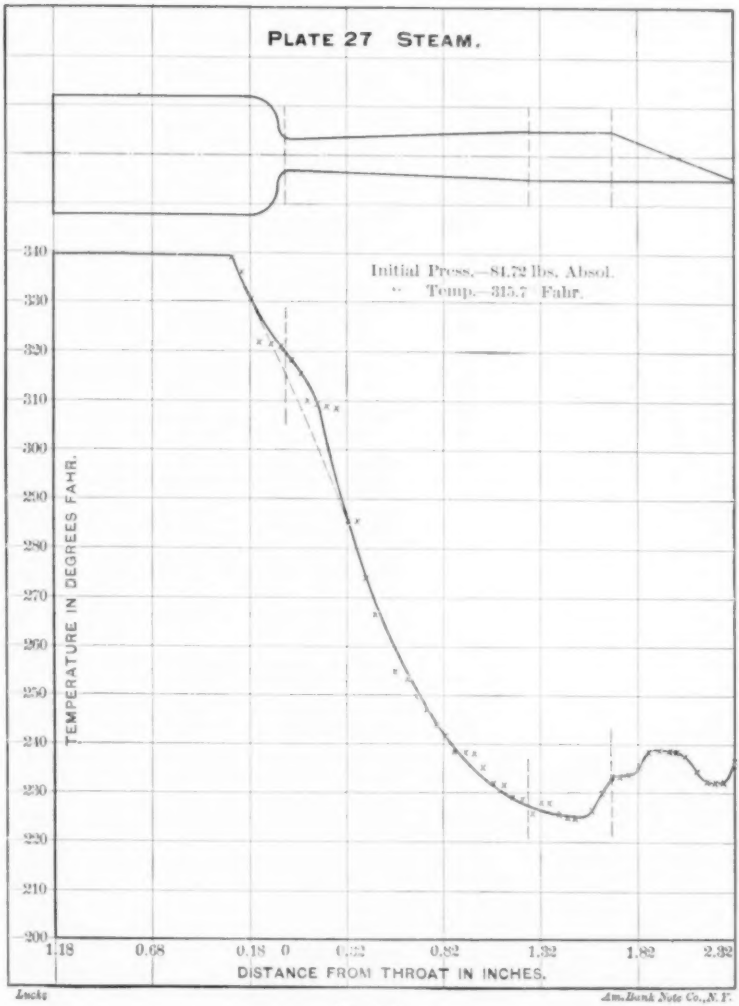
PLATE 27 STEAM.
Initial Press.—81.72 lbs. Absol.
" Temp.—315.7° Fahr.
TEMPERATURE IN DEGREES FAHR.
340
330
320
310
300
290
280
270
260
250
240
230
220
210
200
1.18
0.68
0.18
0
0.32
0.82
1.32
1.82
2.32
DISTANCE FROM THROAT IN INCHES.
Am. Bank Note Co., N.Y.

FIG. 34.

STEAM—INITIAL PRESSURE (80-POUND GAUGE — 94.72 ABSOLUTE).

Distance from Throat in Inches.	Measuring Hole in the Direction of Flow.		Measuring Hole on Side of Searching Tube.		Measuring Hole Opposed to Direction of Flow.	
	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.	Absolute Pressure in ins. of hgt.	Absolute Pressure in lbs. per sq. in.
−0.48	.....	90.72	.....	93.22	.....	93.72
−0.43	.....	90.72	.....	93.22	.....	93.72
−0.38	.....	90.72	.....	93.22	.....	93.72
−0.33	.....	90.72	.....	93.22	.....	93.72
−0.28	.....	90.72	.....	93.22	.....	93.72
−0.23	.....	90.72	.....	93.22	.....	93.72
−0.18	.....	90.72	.....	93.22	.....	93.72
−0.13	.....	90.22	.....	93.22	.....	93.72
−0.08	.....	88.72	.....	92.72	.....	92.72
−0.03	.....	86.72	.....	91.72	.....	92.72
+0.02	.....	75.22	.....	89.72	.....	92.72
0.07	31.75	15.55	.....	81.22	.....	92.72
0.12	30.85	15.11	.....	59.22	.....	92.72
0.17	30.65	15.01	.....	50.72	.....	92.72
0.22	29.65	14.52	.....	44.72	.....	92.72
0.27	29.45	14.23	.....	39.72	.....	92.72
0.32	29.05	14.23	.....	34.72	.....	92.72
0.37	28.05	13.74	.....	29.72	.....	92.22
0.42	25.05	10.26	.....	27.22	.....	91.22
0.47	20.85	10.20	.....	23.22	.....	90.72
0.52	20.65	10.10	.....	21.72	.....	90.22
0.57	19.85	9.71	.....	20.72	.....	89.22
0.62	19.25	9.42	35.65	17.47	.....	87.22
0.67	18.85	9.22	33.55	16.44	.....	83.22
0.72	17.95	8.78	31.05	15.21	.....	82.72
0.77	17.25	8.44	29.95	14.67	.....	83.22
0.82	16.25	7.94	29.35	14.38	.....	82.22
0.87	15.85	7.75	28.15	13.79	.....	82.22
0.92	15.25	7.45	27.55	13.49	.....	80.72
0.97	14.75	7.21	26.65	13.05	.....	78.22
1.02	14.05	6.86	25.65	12.56	.....	74.72
1.07	13.45	6.57	24.95	12.22	.....	75.72
1.12	12.85	6.27	24.05	11.77	.....	75.72
1.17	12.45	6.08	23.25	11.38	.....	73.72
1.22	11.75	5.73	22.75	11.14	.....	72.22
1.27	11.55	5.64	22.15	10.84	.....	72.22
1.32	11.25	5.49	21.55	10.55	.....	62.72
1.37	11.65	5.69	21.25	10.40	.....	76.72
1.42	12.35	6.03	21.35	10.45	.....	71.72
1.47	13.55	6.62	22.25	10.89	.....	68.72
1.52	14.45	7.06	22.45	10.99	.....	68.22
1.57	16.15	7.90	22.65	11.09	.....	64.22
1.62	18.95	9.27	22.65	11.09	.....	67.22
1.67	20.55	10.06	23.25	11.38	.....	66.22
1.72	22.95	11.23	26.65	13.05	.....	61.22
1.77	24.05	11.77	30.35	14.87	.....	55.22
1.82	23.95	11.72	30.25	14.82	.....	57.22
1.87	24.45	11.97	30.15	14.77	.....	59.72
1.92	25.35	12.41	29.25	14.62	.....	60.72
1.97	26.15	12.81	29.25	14.33	.....	61.72
2.02	26.45	12.95	29.65	14.52	.....	60.72
2.07	26.85	13.15	29.95	14.67	.....	60.22
2.12	27.05	13.25	29.25	14.33	.....	60.22
2.17	27.35	13.39	29.15	14.28	.....	60.72
2.22	27.75	13.59	29.35	14.38	.....	63.72
2.27	28.55	13.98	28.55	13.98	.....	63.72
2.32	29.05	14.23	28.25	13.84	.....	60.72

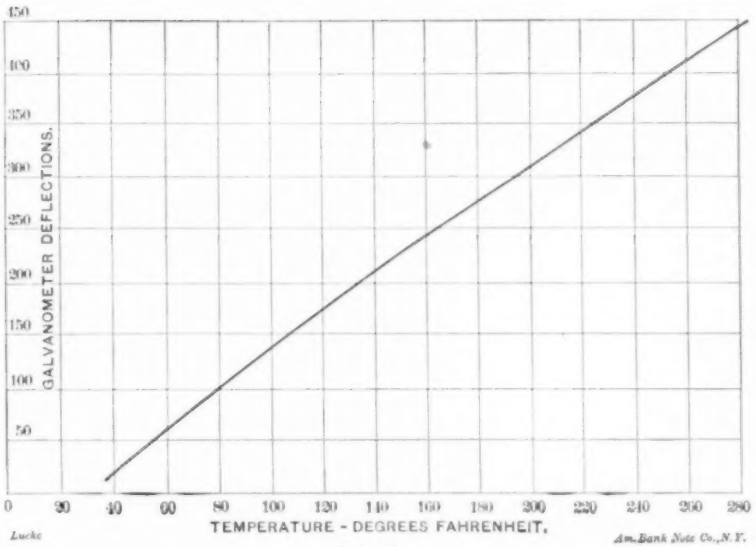

FIG. 35.

DE LAVAL NOZZLE.

NON-CONDENSING

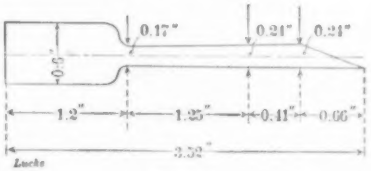

FIG. 36.

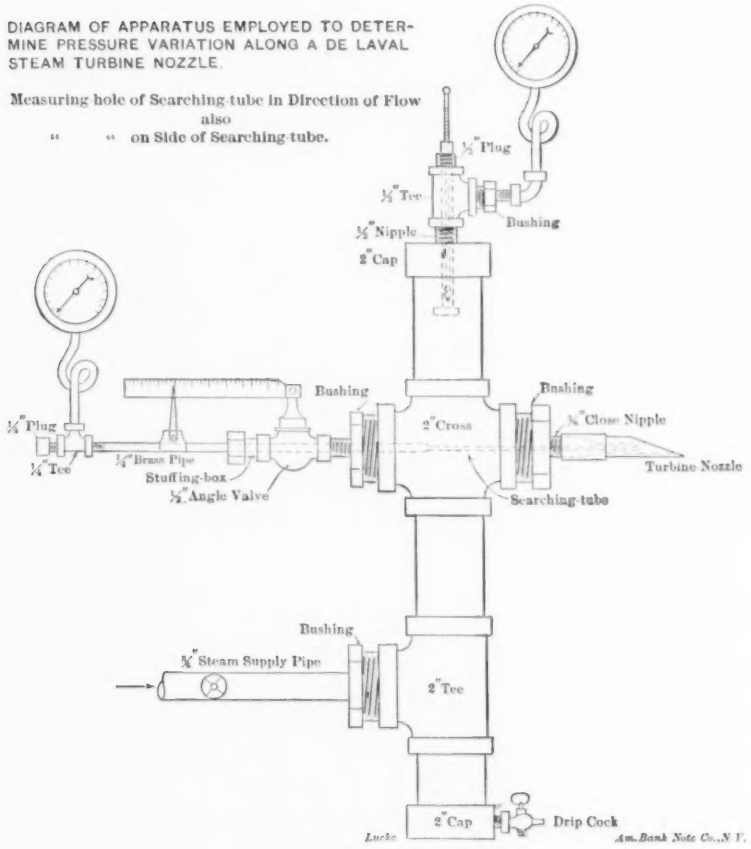
DIAGRAM OF APPARATUS EMPLOYED TO DETERMINE PRESSURE VARIATION ALONG A DE LAVAL STEAM TURBINE NOZZLE.
Measuring hole of Searching-tube in Direction of Flow
also
" " on Side of Searching-tube.
½" Plug
½" Tee
Bushing
½" Nipple
2" Cap
¼" Plug
¼" Tee
¼" Brass Pipe
Stuffing-box
½" Angle Valve
Bushing
2" Cross
Bushing
¾" Close Nipple
Turbine Nozzle
Searching-tube
Bushing
¾" Steam Supply Pipe
2" Tee
2" Cap
Drip Cock
Lucke
Am. Bank Note Co., N. Y.

FIG. 37.

AIR—TEMPERATURES (IN DEGREES FAHRENHEIT).

Distance from Throat in Inches.	Initial Pressures (in Pounds Absolute).					
	54.72	64.72	74.72	84.72	94.72	104.72
	Initial Temperature (in Degrees Fahrenheit).					
	88.0	99.0	99.0	99.0	99.0	99.0
−0.53	88.75	97.0	99.0	100.25	97.75	99.0
−0.48	88.75	97.0	99.0	100.25	97.75	99.0
−0.43	88.75	97.0	99.0	100.25	97.75	99.0
−0.38	88.75	97.0	99.0	100.25	96.75	99.0
−0.33	88.75	97.0	99.0	100.25	96.75	99.0
−0.28	88.75	96.75	99.0	100.25	96.75	99.0
−0.23	88.75	96.75	99.5	100.25	96.75	98.5
−0.18	88.75	96.5	99.5	99.75	96.75	98.5
−0.13	88.0	96.5	99.5	99.5	96.75	98.5
−0.08	84.5	93.0	97.0	95.0	96.0	95.5
−0.03	80.5	90.0	94.5	90.5	92.0	93.5
+0.02	78.0	88.0	91.5	89.0	90.0	91.0
0.07	76.5	88.0	90.5	89.0	86.5	87.5
0.12	74.5	83.25	86.0	85.5	86.0	87.5
0.17	74.0	83.5	85.5	84.0	86.0	87.0
0.22	73.75	83.0	85.0	83.5	85.5	86.5
0.27	73.0	82.5	83.5	81.5	84.5	83.0
0.32	71.0	82.5	82.0	82.0	84.5	85.0
0.37	71.0	82.0	81.0	81.5	78.5	85.0
0.42	70.0	81.0	80.5	81.0	78.0	82.0
0.47	69.0	81.5	80.0	80.75	77.5	80.0
0.52	69.5	82.0	80.0	78.0	77.5	80.0
0.57	69.0	78.5	79.5	76.5	77.5	78.5
0.62	68.5	77.0	76.0	76.0	75.0	78.5
0.67	67.0	78.0	75.5	76.0	75.0	77.75
0.72	66.0	77.0	75.0	75.5	73.0	77.75
0.77	64.75	75.0	75.0	75.0	72.0	77.75
0.82	64.5	72.0	74.5	75.0	72.0	75.0
0.87	64.5	72.0	74.5	75.0	71.5	75.0
0.92	64.5	72.0	73.0	75.0	71.0	72.5
0.97	64.0	72.0	73.0	76.0	68.0	72.5
1.02	64.0	69.0	73.0	73.5	67.5	71.5
1.07	64.0	63.5	72.0	73.5	67.0	71.0
1.12	63.5	60.5	70.0	72.5	67.0	71.0
1.17	61.5	54.0	72.0	72.0	67.0	70.5
1.22	62.25	53.75	71.5	70.75	67.0	71.0
1.27	62.25	51.5	69.0	70.75	66.5	70.0
1.32	63.5	49.5	68.5	70.75	66.5	69.0
1.37	63.5	48.5	68.5	68.0	66.5	70.0
1.42	62.25	48.0	68.5	69.0	66.5	71.0
1.47	62.25	45.5	68.5	69.0	66.0	70.5
1.52	62.25	45.5	69.0	69.0	65.0	70.5
1.57	64.0	43.0	69.0	69.0	66.0	70.5
1.62	64.5	43.5	68.5	67.5	66.0	70.5
1.67	64.5	43.5	72.0	67.5	66.0	70.5
1.72	64.5	45.0	72.0	70.0	66.0	70.5
1.77	64.75	43.5	73.5	71.0	66.0	70.0
1.82	64.75	43.5	72.0	71.0	66.0	70.0
1.87	66.0	43.5	73.25	70.0	69.5	70.5
1.92	67.0	43.5	73.5	70.0	67.0	70.5
1.97	68.0	44.0	72.0	67.75	67.0	70.5
2.02	68.5	44.5	73.0	67.5	68.0	73.0
2.07	68.5	42.5	72.75	64.75	68.0	71.0
2.12	69.0	41.0	71.5	63.5	67.0	70.5
2.17	69.2	43.5	73.0	61.0	66.5	73.0
2.22	69.5	42.5	73.0	61.0	66.0	71.0
2.27	69.2	44.0	73.0	62.0	65.5	70.5
2.32	73.0	44.5	72.0	64.5	65.0	69.5

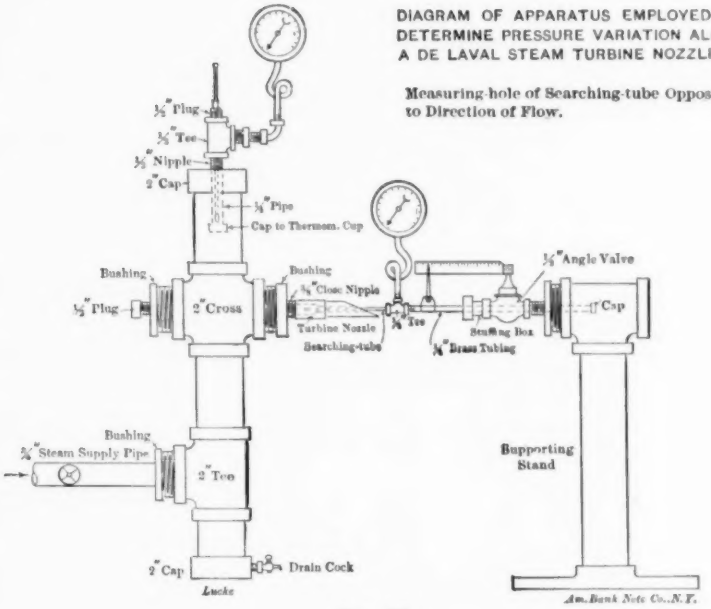
DIAGRAM OF APPARATUS EMPLOYED TO DETERMINE PRESSURE VARIATION ALONG A DE LAVAL STEAM TURBINE NOZZLE.
Measuring-hole of Searching-tube Opposed to Direction of Flow.
⅛" Plug
⅛" Tee
⅛" Nipple
2" Cap
⅛" Pipe
Cap to Thermom. Cup
Bushing
½" Plug
2" Cross
Bushing
⅛" Close Nipple
Turbine Nozzle
Searching-tube
⅛" Tee
⅛" Angle Valve
Stuffing Box
Cap
¼" Brass Tubing
Bushing
⅜" Steam Supply Pipe
2" Tee
Supporting Stand
2" Cap
Drain Cock
Lucke
Am. Bank Note Co., N.Y.

FIG. 38.

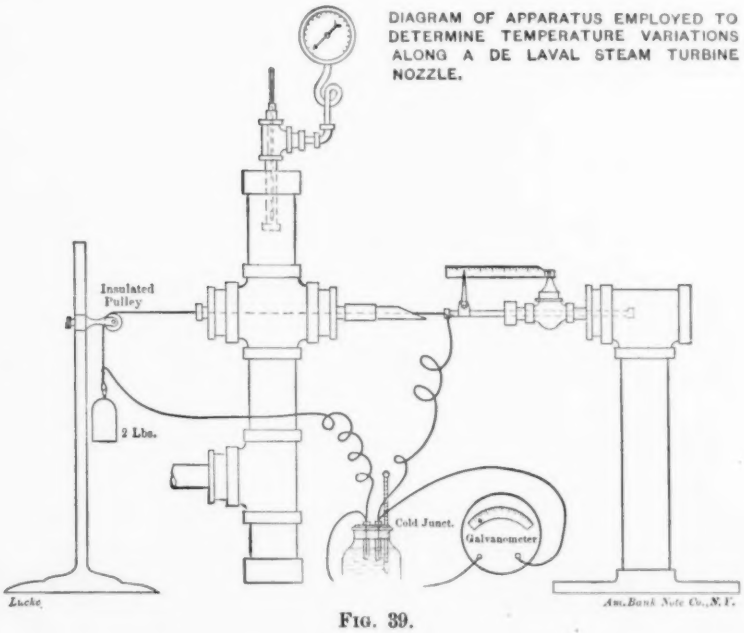
DIAGRAM OF APPARATUS EMPLOYED TO DETERMINE TEMPERATURE VARIATIONS ALONG A DE LAVAL STEAM TURBINE NOZZLE.
Insulated Pulley
2 Lbs.
Cold Junct.
Galvanometer
Lucke
Am. Bank Note Co., N.Y.

FIG. 39.

STEAM—TEMPERATURES (IN DEGREES FAHRENHEIT).

Distance from the Throat in Inches.	Initial Pressures (in Pounds Absolute).			
	54.72	64.72	74.72	84.72
	Temperature of Saturated Steam for Initial Pressure (in Degrees Fahr.)			
	286.6	297.5	307.0	315.7
−0.53	315.5	323.0	329.3	339.7
−0.48	315.5	323.0	329.3	339.7
−0.43	315.5	323.0	329.3	339.7
−0.38	315.0	323.0	329.3	339.7
−0.33	315.0	323.0	329.3	339.7
−0.28	315.0	323.0	329.3	339.7
−0.23	314.0	323.0	329.3	336.7
−0.18	309.0	320.0	320.3	330.7
−0.13	303.0	314.0	308.3	322.7
−0.08	306.0	311.5	296.3	322.7
−0.03	303.0	311.0	290.3	321.7
+0.02	300.0	308.0	287.3	318.7
0.07	300.0	305.0	283.8	315.7
0.12	297.0	304.5	278.8	310.2
0.17	297.0	302.0	273.3	309.7
0.22	296.0	299.0	272.8	309.7
0.27	294.0	302.0	267.3	308.7
0.32	279.0	290.0	264 8	285.7
0.37	288.0	287.0	259.8	285 7
0.42	279.0	281.5	257.8	274.2
0.47	273.0	278.0	253.8	266.7
0.52	269.0	275.5	251.8	259.7
0.57	269.0	266.0	248.3	255.2
0.62	261.5	262.5	244.3	253.7
0.67	260.0	256.5	240.3	249.7
0.72	255.5	251.0	237.3	247.7
0.77	252.0	247.0	234.3	244.2
0.82	249.5	245.0	234.3	241.7
0.87	246.5	243.0	231.3	238.7
0.92	244.0	240.0	228 3	238.7
0.97	241.0	233.0	224.8	238.7
1.02	235.0	231.0	224.8	235 7
1.07	235.0	227.5	222.3	232.7
1.12	232.0	226.0	221.8	231.7
1.17	231.5	226.0	219 3	229 7
1.22	231.5	226.0	219.3	229.7
1.27	228.5	226 0	221.3	226 2
1.32	232.0	226.0	219.3	228.2
1.37	228.5	227.5	219.3	228.2
1.42	223 0	221.5	218.8	226.2
1.47	219.0	220.0	219 3	225.7
1.52	221.0	217.0	222.8	225.7
1.57	226.5	218.0	224.8	226.2
1.62	238.0	222.0	227.5	229.7
1.67	235.0	231.0	228.3	232.7
1.72	235.0	235.5	231.3	232.7
1.77	235.0	239.0	237.3	232.7
1.82	238 0	237.0	234.3	234.7
1.87	238.0	237.0	234.3	238.7
1.92	238.0	239.0	231.3	238.7
1.97	246.0	237.0	231.3	238.7
2.02	246.0	245.5	231.3	238.7
2.07	244.0	243.0	231.3	237 7
2.12	244.0	234.0	231 3	234.7
2.17	244.0	234.0	231.3	232.7
2.22	245.0	234.0	231.3	232.7
2.27	244.0	234 0	231.3	232.7
2.32	244.0	234.0	231.3	235.7

DISCUSSION.

Mr. R. P. Bolton.—I have but a few words to say with reference to this paper, in the hope that the subject matter will still further be investigated. It contains matter of value in connection with the subject of heating, which is not apparent on the face of the paper. Much of the steam taken for heating apparatus is passed through reducing valves, being emitted by such valves through very small orifices, and thus partakes of the nature of a jet, passing from a high pressure to a low pressure. The curious effect has been several times observed in actual work that the amount of steam used from an exhaust steam apparatus may be sufficient to heat a certain capacity in a building, but that if that apparatus is stopped, and live steam is taken direct from the boiler through the reducing valve for the same work, more boiler capacity has to be brought into operation. Now there must be something behind that effect, and it appears to me that this paper gives a line of thought which will lead toward a solution of what has hitherto been a mystery to heating engineers.

Prof. F. R. Hutton.—I think it will also add greatly to the interest of this discussion if attention is called to the fact that the research looks towards a possible solution of the gas turbine. One of the great difficulties in the attempt to apply the turbine method to a gas engine has been that the drop of temperature in free expansion in the nozzle does not take the heat away, but only pressure. The trouble is that the exhaust goes out too hot to have been of any use.

No. 1060.*

THE POWER PLANT OF THE TALL OFFICE BUILDING.

A COMPARISON BETWEEN COMPOUND AND FOUR-VALVE SIMPLE ENGINES RUNNING NON-CONDENSING.

BY S. H. BUNNELL, WATERTOWN, N. Y.
(Member of the Society.)

1. The limit of excellence in steam engine economy stands at about one horse-power produced per pound of coal burned per hour. This duty is only reached by large and thoroughly well-built engines under the most favorable conditions of uniform load, high steam pressure and good vacuum in condenser. Between this remarkable figure and the actual everyday performances of the high-speed engines necessarily used in the restricted spaces of the engine rooms of large city buildings there is a very wide gap. The problem before the designers of power plants of the latter class is to make the best use of steam received at moderately high pressure, and delivered into heating systems at a pressure more or less above the atmosphere, with the advantage of atmospheric exhaust only during some five months of the year. The engines using steam under these conditions must develop powers widely varying from small percentages of the full load up to the greatest overload capacity which may be required when nearly all of the elevators of the building happen to be taking their several maximum amounts of current. Together with these requirements it is demanded that the regulation of the engines shall be practically perfect, as measured by the production of incandescent electric lights without perceptible fluctuation.

2. For these conditions high-speed engines are almost invariably

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

required and selected. These engines are of three principal types: the simple single-cylinder engine with single steam valves, the single-cylinder engine provided with four valves of Corliss or slide type, thereby separating the inlet from the exhaust and giving practically the economy of the regular slow-speed engine of Corliss type, and the non-condensing compound engine, usually provided

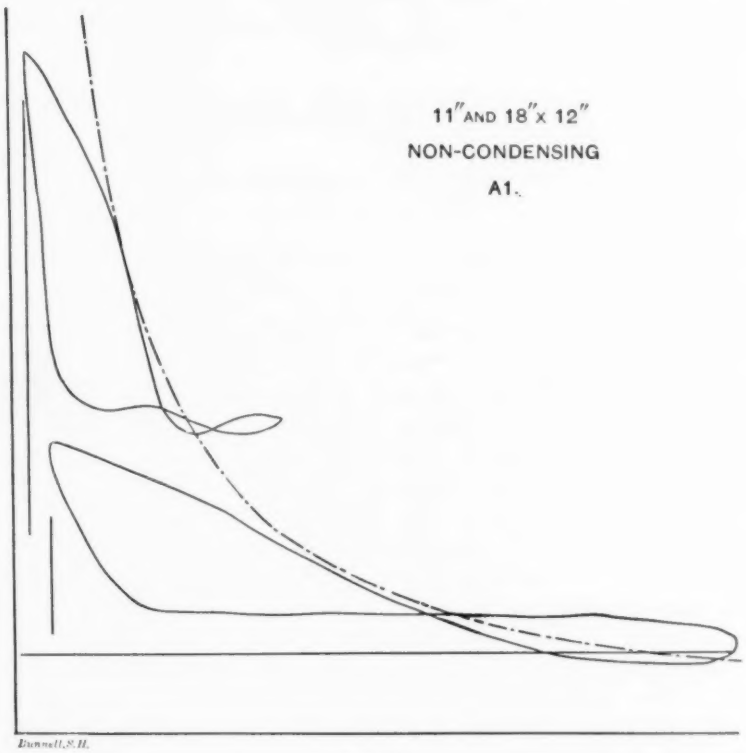

FIG 40.

with a single valve for each cylinder, but sometimes having separate steam and exhaust valves on one or both cylinders. In most cases these engines seem to be designed to handle the maximum loads of the generators directly connected to them, without much regard to their probable performance in the matter of economy and use of steam when running under loads much below the maximum. It is evident that with exhaust back pressure not less than atmospheric pressure, and most of the time considerably more

than this, the number of possible expansions or the ratio of initial to terminal steam pressure is limited to a figure which is not unsuitable for simple engines. Since steam expansion cannot be economically carried below terminal pressure without a waste of developed power shown by the formation of a negative loop in the indicator diagram, the minimum power economically developed

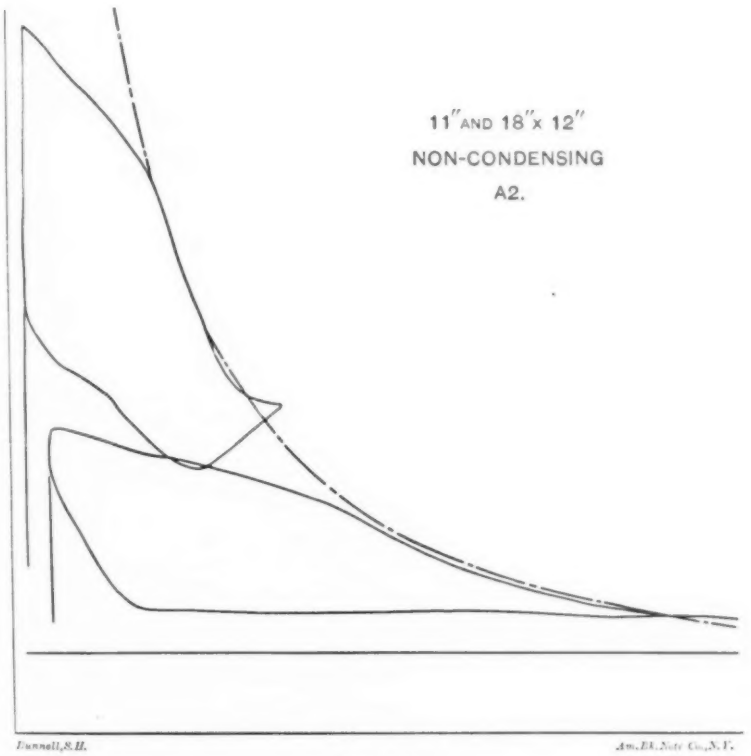

FIG. 41.

by the non-condensing engine is that due to the mean effective pressure consequent upon making the terminal pressure equal to the back exhaust pressure. From this minimum the power can be increased without limit until the engine is taking steam full stroke and operating without expansion. A division of the possible diagram into two separate diagrams by compounding is at the expense of more or less of a gap between the two diagrams thus produced, due to valve friction, condensation in receiver

and other losses, the only compensation for which is the decreased range of temperature between inlet and exhaust in the separate cylinders, compared with the range existing in the single-cylinder engine with the same inlet and exhaust pressure. As a matter of fact, indicator cards and tests from compound and simple non-condensing engines under the conditions existing in the power

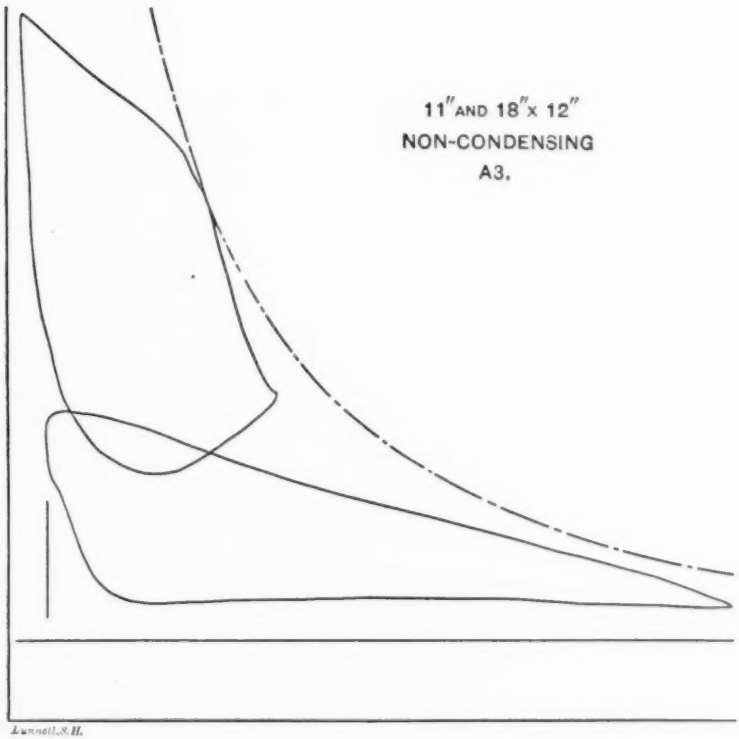

FIG. 42.

plants of city buildings show that the employment of compound engines for this service often gives results inferior to the performance of single-cylinder engines with better valve gear and costing considerably less to construct.

3. Some time ago Mr. Isaac D. Parsons made a careful study of the conditions of operation of a number of power plants of the class under consideration. His figures were selected as the best and most definite obtainable by an investigation which covered

very many buildings and plants differing widely in details, sizes and conditions. Data obtained from seventeen plants whose records were accurate and complete were tabulated in convenient form for determining the actual cost of current as compared with the rates charged by central stations. The items making up the cost of current were taken to be labor, coal and handling of ashes,

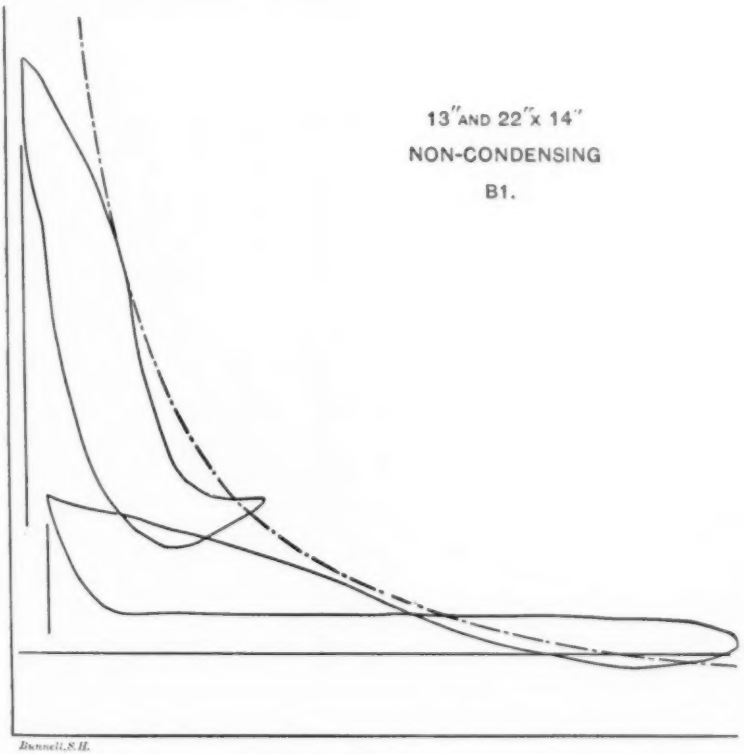

FIG. 43.

water, lamps, oil and supplies, repairs, central station service where used for periods of minimum consumption to allow of shutting down plant, and interest and depreciation at six per cent. per annum on original cost of plants. Roughly these figures average labor $\frac{1}{3}$, coal $\frac{1}{3}$, and interest and depreciation $\frac{1}{10}$ to $\frac{1}{3}$, the sum giving the cost of power being made up by the items of minor expense. The amount paid for labor does not depend closely upon the details of the plant installed, except that simplicity of

machinery would tend to decrease the cost of labor necessary to properly operate it. A saving in the cost of current would, therefore, be necessarily made chiefly by decreasing the consumption of coal and the first cost and expense of repairs to the machinery.

4. Table 1 is adapted from Mr. Parsons' figures which were published in the "Engineering Magazine" of January and February, 1902. The last column gives the ratio of the maximum momentary load which might be caused by the simultaneous start-

TABLE I.

Number.	Kind of Building.	Number of electric elevators and maximum kw. current required.	Minimum lighting load.	Duration of minimum load.	Generators.		Average per cent. total capacity developed.	Steam pressure, pounds.	Back pressure, pounds.	Lb. coal per KW. hour including steam pump and any live steam for heating.	Ratio maximum day load to minimum day load.
					Number.	Total capacity.					
		KW.	*KW.*	*Hrs.*		*KW.*					
1	Office b'l'd'g..	4–100	11	8	4	260	.55	90	0	15.5	100–10
2	" " ..	10–250	11	8	4	575	.75	125	0	10.	100– 4
3	" " ..	6–162	44	10	5	475	.75	115	1– 3	9.1	100–21
4	" " ..	8–240	..	..	3	300	.90	100	4	8.2	.....
5	" " ..	3– 60	..	..	1	40	.65	95	1– 5	11.4	.....
6	Power " ..	15–300	..	..	3	300	.90	90	3– 9	11.	.....
7	" " ..	4– 80	..	..	3	260	.75	100	2– 5	13.5	.....
8	" " ..	4–100	..	..	3	90	.60	100	7–10	14.4	.....
9	Dept. Store...	none.	25	6	3	288	.75	90	....	....	.....
10	Apartment...	1– 15	3	8	2	45	.48	90	5	15.5	100–17
11	" ...	2– 40	2	8	2	60	.20	100	2– 5	25.1	100– 5
12	" ...	3– 75	..	..	3	175	.60	125	3– 5	18.4	.....
13	Hotel.....	none.	11	17	2	66	.50	90	4	13.5	.....
14	"	2– 30	28	..	2	80	.67	100	3– 6	12.4	100–48
15	Club.....	none.	8	9	1	20	.40	80	6	26.5	.....
16	"	1– 20	3	12	3	130	.40	90	3– 5	11.1	100–13
17	"	none.	..	..	2	50	.75	80	3– 5	16.6	.....

ing of all the elevators to the minimum lighting load existing during the hours of bright daylight. The extreme ratio shown in any of these buildings is that of No. 11, where the minimum load is (supposing the operation of both elevators during daytime) only 4 per cent. of the maximum capacity which may be suddenly called for. This, however, is an extreme figure; from ten to twenty per cent. seems more representative of the minimum load required to be carried by the power plant. It is evident that the mean of the load carried throughout the day by any of these engines would be about one-half of the full load, and that the

engines should be designed for best economy at some point considerably below their maximum power, and to operate with reasonable efficiency over a wide range of load. While this is true of all engines of isolated plants for furnishing current to buildings, it is especially the case in the larger plants, since the steady lighting load of office buildings is considerably below the maximum, and the maximum load is only required for an hour or two in the afternoon during the winter months.

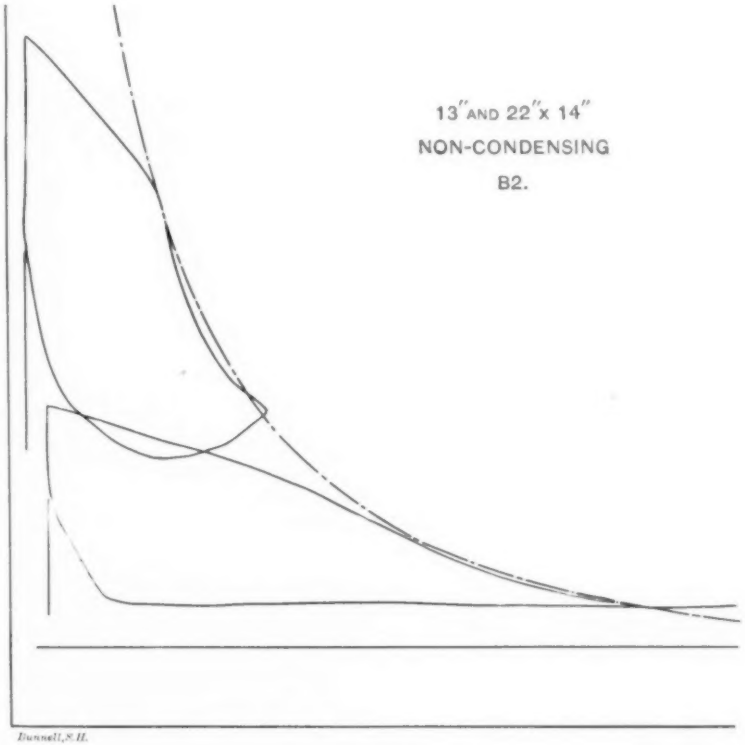

FIG. 44.

5. The accompanying diagrams have been laid out from actual indicator cards produced by Watertown engines of the usual high-speed non-condensing type. These cards are representative of many taken from the engines under varying conditions of pressures and loads, but for the purpose of this paper have been selected to show the performances of these engines under a boiler

pressure of 110 pounds by gauge, and exhausting against a back pressure of five pounds in the exhaust pipe, or about seven pounds in the cylinder. *A* is a compound engine with tandem cylinders 11 and 18 by 12, each having a single steam valve driven from the same governor. The throttling due to the extremely small valve openings at light loads and the wasteful loops produced in

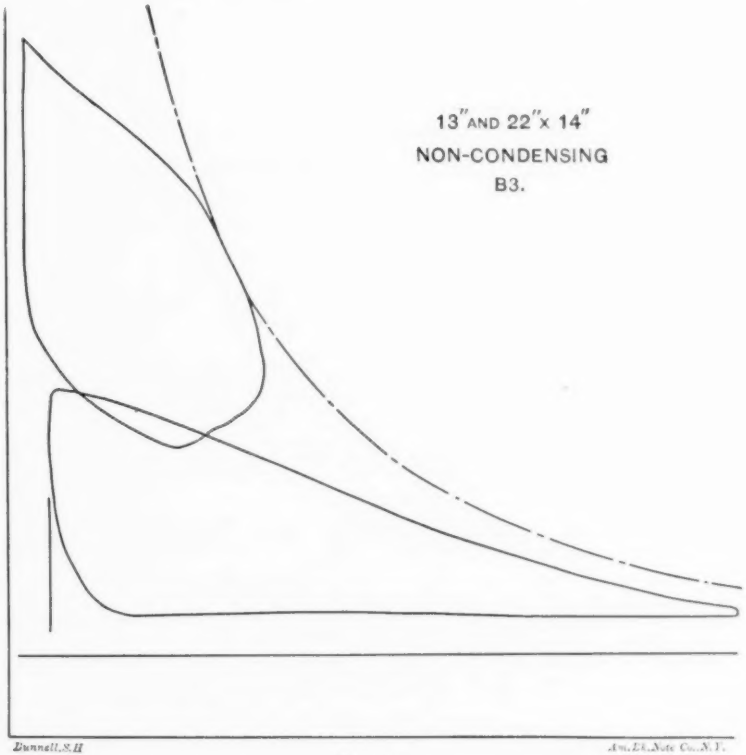

FIG. 45.

both cylinders under such conditions are features of the diagrams from this engine. *B* is a tandem-compound with single-valve cylinders 13 and 22 by 14 with the low-pressure valve driven by the usual eccentric, the high-pressure valve only being connected to the governor. *C* is a 20 by 18 single cylinder four-valve engine with valves placed close to the bore of the cylinder to reduce clearance, the exhaust valves driven by a fixed eccentric and the inlet controlled by the governor. The cards from engine

C and the low-pressure cylinder of engine *B*, therefore, show uniform release and compression. The diagrams are in four sets. No. 1 shows considerable underload for all three engines. No. 2 shows the most economical loads for the compounds and a proper load for the four-valve to give satisfactory duty. No. 3 shows the maximum loads for the compounds, which have already passed

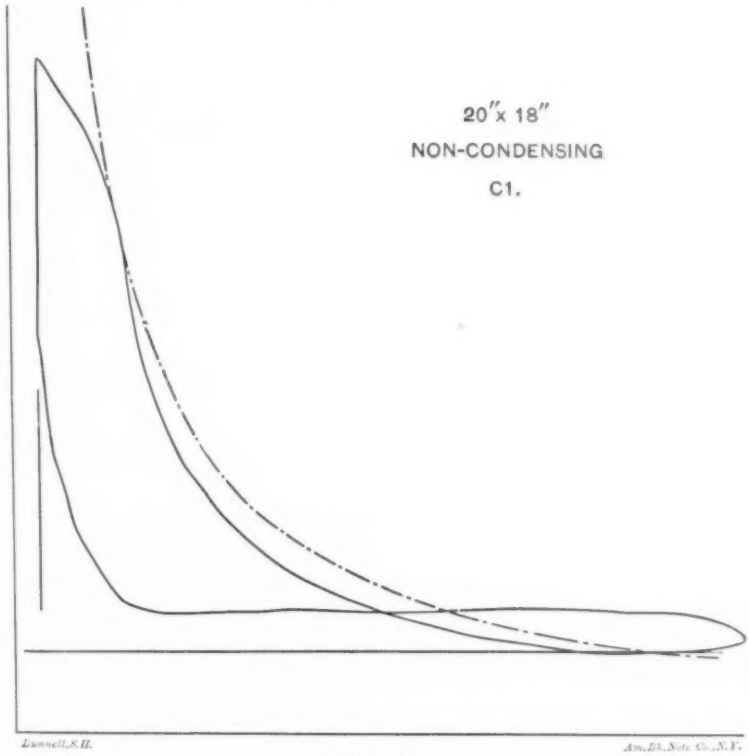

FIG. 46.

their points of best efficiency, while the same amount of steam in the four-valve gives much better results. No. 4 shows the four-valve with a load about 160 per cent. of the maximum carried by the compounds, and with substantially the same water consumption.

6. It is apparent that the four-valve engine gives very much better steam distribution and economy than the compounds. The reëvaporation during expansion even at light loads is not great,

and does not indicate any such loss of efficiency as is due to the large gaps of the compound diagrams. For the purpose of comparison, Table II. shows the mean effective pressure from the cards as given, and also, with but two pounds back pressure in the cylinders, the corresponding water consumption shown by the

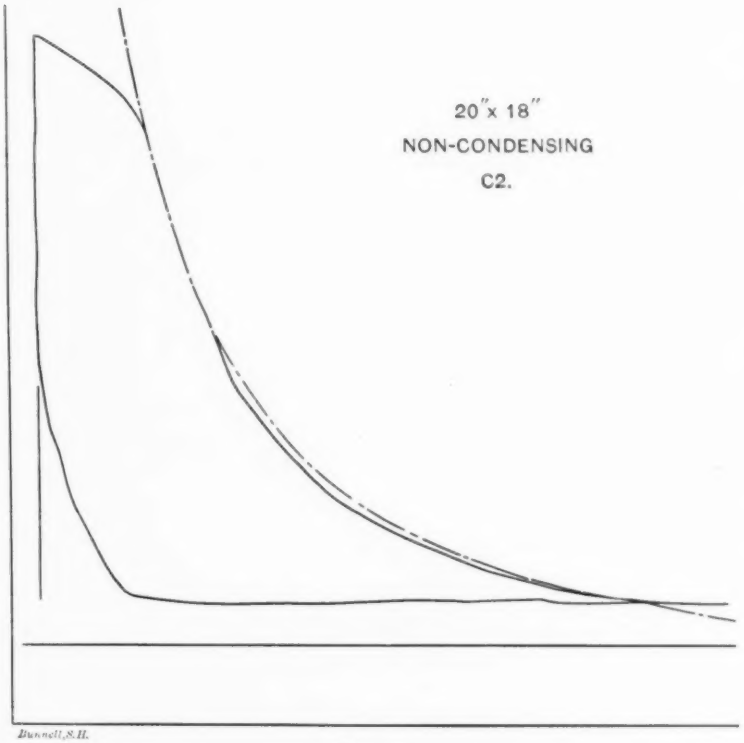

FIG. 47.

diagrams, and the probable percentage of steam wasted by cylinder condensation, with the actual water consumption corresponding. The percentages of load shown by each set of figures, compared with the full load capacity shown by set No. 3, is given in this table.

TABLE II.

Diagram.	5 Lb. Back Pressure.					Atmospheric Exhaust.				
	M.E.P.	Load in per cent. of Diagram 3.	Water per h. p. hour.			M.E.P.	Load in per cent. of Diagram 3.	Water per h. p. hour.		
			Pounds Indicated.	Assumed per cent. Condensation.	Pounds Est. Actual.			Pounds Indicated.	Assumed per cent. Condensation.	Pounds Est. Actual.
		p.c.		*p.c.*					*p.c.*	
A—1.....	10.1	31	40.9	20	51.0	15.1	40	27.3	20	34.1
B—1.....	11.0	33	36.2	20	45.3	16.	42	24.9	20	31.1
C—1.....	11.93	31	33.5	28	46.5	16.93	39	23.5	28	32.7
A—2.....	24.7	75	24.5	15	28.9	29.7	79	20.4	15	24.0
B—2.....	25.1	76	23.6	15	27.8	30.1	79	19.6	15	23.1
C—2	25.9	67	22.3	25	29.8	30.9	71	18.7	26	25.3
A—3.....	32.8	100	26.8	12	30.5	37.8	100	23.2	12	26.4
B—3.....	33.2	100	26.1	12	29.7	38.2	100	22.6	12	25.7
C—3.....	38.4	100	21.8	22	27.9	43.4	100	19.3	24	25.4
C—4.....	53.0	138	22.0	18	26.8	58	134	20.2	20	25.3

Note.—M.E.P. of compound engines is referred to low pressure cylinder area.

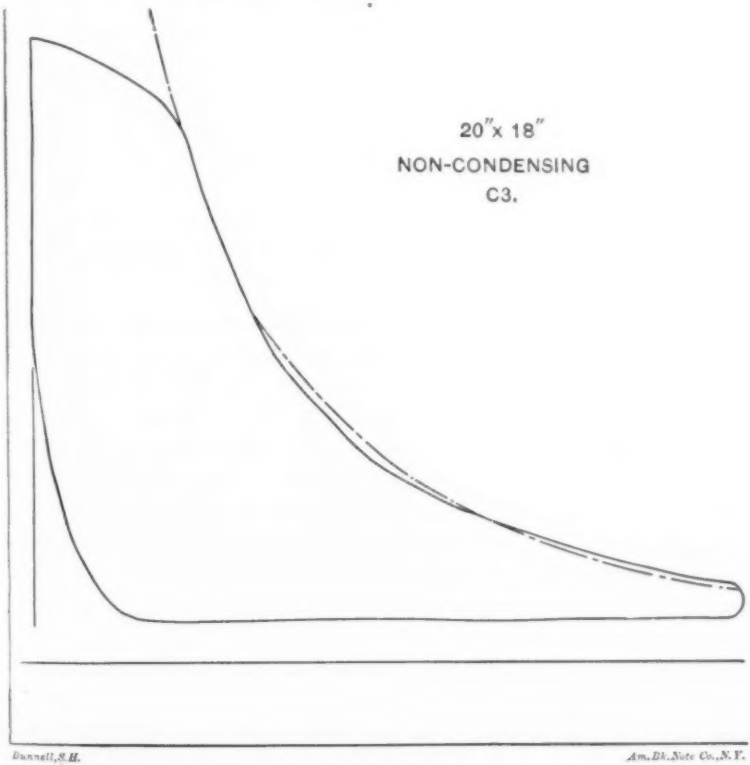

Fig. 48.

7. These figures indicate that the single-cylinder four-valve engines has a far greater range of capacity with reasonable economy than the non-condensing compound engine and for the actual conditions of service in power plants of large buildings is much to be preferred. It would seem very desirable that engineers planning power installations of this character should make

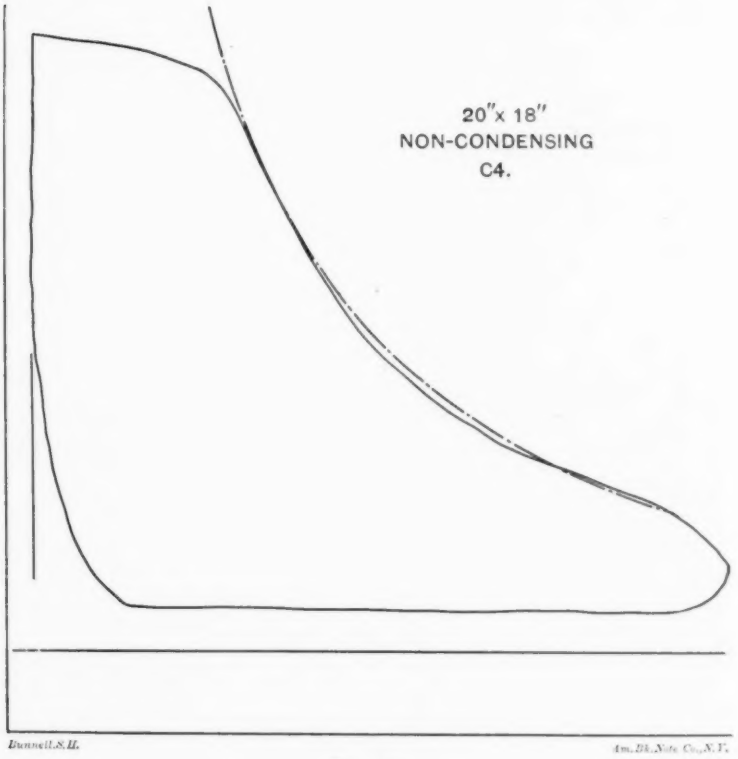

FIG. 49.

a careful study of the probable actual running conditions under which current must be furnished so as to select engines designed to give the best economy under actual conditions instead of as is too frequently done letting contracts on the basis of the best load performance guaranteed, in spite of the fact that, in buildings where most of the electric current generated is used for power by the various auxiliary machines and the elevators, the full-load capacity is only infrequently and momentarily required.

DISCUSSION.

Mr. J. B. Stanwood.—Using the analysis described in my paper, I have taken Mr. Bunnell's Table II with engines under *atmospheric exhaust*, and have calculated the "Cylinder Efficiencies."

As 16.7 pounds is the theoretical minimum water rate for 110 pounds boiler pressure by gauge (see my paper, Table XI), then the ratio of 16.7 to the expected water rates of Mr. Bunnell gives for each example three different cylinder efficiencies.

In the diagram, Fig. 50, are plotted these values for B and C engines only (one compound and one simple), and it shows plainly the advantage of capacity possessed by the simple engine, and of efficiency by the compound engine.

Mr. Bunnell assumes a condensation loss of 20 per cent. with equal light loads for both types of compound engines. As the engine with governor controlling the valve gear of the high pressure cylinder only has a greater temperature range at light load than the other, it would probably be found in practice, all things equal, to have a greater initial condensation than the engine in which the valves of both cylinders are under governor control.

With the simple engine diagrams "C_1" and "C_2," the cut-offs by measurement are 0.10 and 0.14. The assumed per cents. of condensation are 28 and 26. I do not know of any records of tests showing so low a percentage for these early cut-offs. In Mr. Barrus' book on "Engine Tests," the range for 0.10 cut-off is from 33 to over 40 per cent., and for 0.15 cut-off, from 30 to 35 per cent. Assuming in Mr. Bunnell's cases the condensation to be as low as 35 and 30 per cent. respectively, there will then be found a water rate of 36.1 instead of 32.7; and 26.8 instead of 25.3 respectively, corresponding cylinder efficiencies being indicated by the dotted line on the diagram.

If Mr. Bunnell has any tests in which this low record of condensation for these early cut-offs has been obtained, it will be of value to present them to the Society.

The boiler pressure used is too low in order to get the best results from compound engines of the cylinder ratio as given, that is, 2.6 to 1. For high back pressures and 110 pounds boiler pressure, I should advise a cylinder ratio of about 2 to 1 as it

would give capacity with only a slight reduction in economy over the larger ratio.

Diagrams taken from a slow motion 11 by 16 by 20-in. cross-compound throttling engine, high pressure cylinder steam jack-

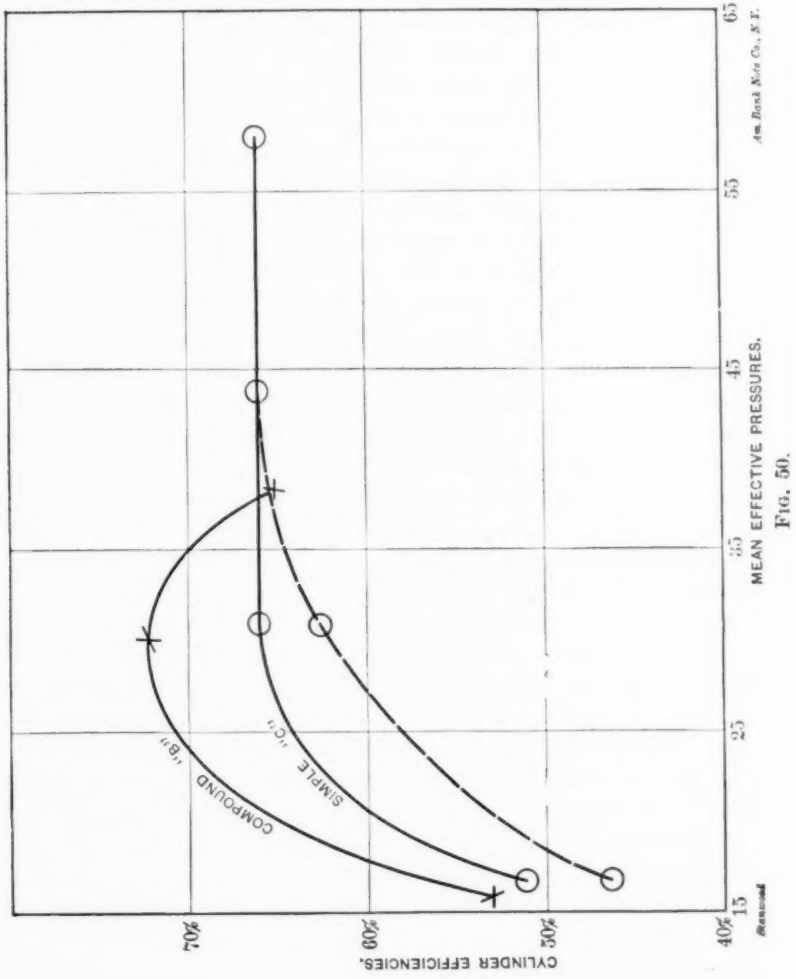

FIG. 50.

eted, has shown a maximum of 49 pounds mean effective pressure, boiler pressure 116 pounds by gauge, cylinder ratio 2.12 to 1, high pressure cut-off 50 per cent.

Incidentally, it may be noted that when engines are working

into a heating system, high economy with light loads is not necessary as usually there is a large demand for exhaust steam.

Mr. F. H. Ball.—This paper is presented to the Society for the announced purpose of making a comparison between the efficiency of two types of engines, particularly under wide changes of load.

In making comparisons of this kind it has heretofore been thought necessary to test the two types that are to be compared, and base the comparison on results shown by these tests. The author of this paper has established a new precedent in dispensing with the formality of tests. He *guesses* at what one type of engine will do, and without even guessing at the other, he is content to merely *assert* that one type is more efficient than the other.

There may be engineering subjects which form an interesting theme for speculation; for instance, a new type of machine proposed to be built, or an untried process, but what excuse can there be for guessing at the efficiency of a small, self-contained engine when it is so simple a matter to test it.

In looking over this paper it is noticeable that the only information regarding the efficiency of either type of engine is found in Table II. on page 169. This table is made in the following manner. First the diagrams are measured to determine the "indicated" water consumption. Then the condensation loss is "estimated," and by combining "indicated" water with "estimated" condensation, the author is able to announce the figures in the final column, which he calls "Pounds estimated actual."

This is a new engineering term. We have all heard of "pounds estimated" and of "pounds actual," but who has ever seen the two in partnership before.

The conclusion reached by this process of estimation is that a four-valve engine with shaft governor is more efficient than a compound engine of the single valve type. Let us see how this conclusion fits the facts obtained by carefully conducted engine tests.

The accompanying chart (Fig. 50A) shows graphically results of a large number of tests reported by George H. Barrus in his book entitled "Engine Tests," published by D. Van Nostrand, 1900.

Each test is represented by a point on this chart, which is located so that the measurements on ordinates indicate the water consumption and on abscissæ the percentage of load during the test. All of the engine tests were run non-condensing and most of them were Corliss engines.

It so happens that the Corliss engine making the best showing

for any simple engine was tested with six loads ranging from 13 per cent. to 1.15 per cent. This enables us to plot the curve *C D* made from test No. 31 on page 126 of Mr. Barrus' book. The engine tested in this case was a 300 horse-power double Corliss engine with 100 pounds steam. The water consumption ranged from 73.63 to 25.39.

In this same book on page 216 test No. 54 gives us the figures obtained from a single valve cross compound engine with 165 pounds steam. The higher pressure here used was no doubt an advantage to the engine, and the comparison with the Corliss engine on the score of efficiency, therefore, is not of much interest, but the curve of efficiency with different loads is interesting, because we are told by the author of this paper that the compound engine is less efficient than the four-valve type with light load.

Curve *A B* represents the figures obtained at loads ranging from 46 per cent to 1.08 per cent. This curve certainly contradicts the conclusion announced in the paper under discussion. It may be interesting to note in this connection the result of a test of a little 75 horse-power single valve compound engine with 100 pounds pressure, non-condensing.

I happened to participate in this test, which was carefully made with a surface condenser, and the results may be accepted with confidence.

The results of this test are represented by circles in the chart and in some cases the test was duplicated as indicated. In this case the test covered almost exactly the same range of load as the test of the Corliss engine and the efficiency is practically the same at every stage of load.

In making this comparison it must be borne in mind:

First. That a 75 horse-power compound is being compared with a 300 horse-power Corliss engine.

Second. That the Corliss engine was one that made the best showing reported in Mr. Barrus' book.

Third. That a short stroke, four-valve engine without releasing valve gear cannot possibly make as good a showing as the average Corliss engine and must therefore fall considerably short of the exceptional performance of this Corliss engine.

The conclusions from this chart are these:

First. That a single valve compound engine is more economical than the best type of Corliss engine of the same horse-power capacity and considerably more economical than any four-valve engine without releasing gear.

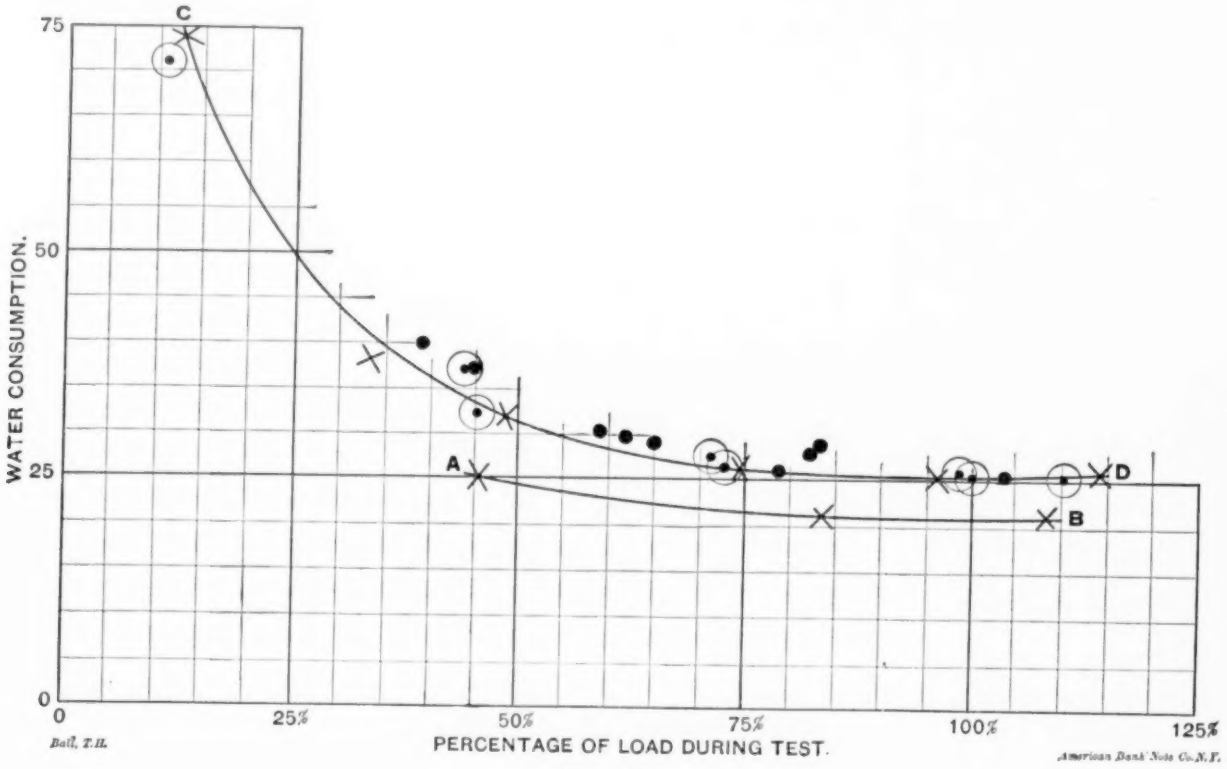
C
75
50
25
0
WATER CONSUMPTION.
A
D
B
0
25%
50%
75%
100%
125%
PERCENTAGE OF LOAD DURING TEST.
Ball, T.H.
American Bank Note Co. N.Y.

FIG. 50A.

Second. That the greater efficiency of the compound engine when compared with the simple four-valve engine is just as noticeable with light loads as with heavy loads.

*Mr. S. H. Bunnell.**—It has been generally assumed, it seems, by architects and engineers in calling for these city engines, that there is no doubt at all about the compound engine being the best at all loads; and it has been our experience that there is some question about that—not at all whether a compound engine can be made more efficient at any load, but whether the types of compound engines ordinarily sold for city building work, with single valves and shaft governors and costing considerably more than simple four-valve engines, are actually as efficient as the latter. Mr. Stanwood's comparison with Table No. II. is a most interesting one. I am glad to accept his judgment as to percentages of condensation, and to note that the performance of the simple engine is still close to that of the compounds. Certainly higher initial pressures will favor the compounds, but present practice limits steam pressures so as to demand lower cylinder ratios and correspondingly reduce the advantages of compounding. A 2 to 1 compound will hardly justify its increased first cost over the simple engine.

Mr. Ball suggests that the heading of the last column is wrong. "Pounds total estimated" will be an improvement and may satisfy his desire for accuracy. His insistence upon accurate scientific tests as a basis for conclusions as to engine efficiency is justifiable—but office building service is too urgent to allow of interference for such purposes, while all of us who build or use engines are well aware that accurate tests of new engines in expert hands give results never realized in commercial operation over long periods.

While the subject under discussion is the relative actual water and coal consumptions of different types of engines in a class of plants where attendance is often not the best, coal and water records only obtainable by searching the books for bills paid, loads widely varying and often very light and tests with scientific accuracy impossible, Mr. Ball offers presumably as his idea of scientific accuracy conclusions drawn from short tests of large and well adjusted engines under good conditions rather than from indicator diagrams from the actual types of engines being considered. In support of his claims he cites a pair of 150 horse-

* Author's Closure, under the Rules.

power Corliss engines under 100 pounds steam in comparison with a cross-compound engine under 165 pounds, the pressure conditions of both engines being unquestionably right for good efficiency over a large range. The total number of possible expansions from 180 to 20 pounds absolute is nine without producing terminal loops, while the engines of the paper are compared at the maximum pressures usual in city buildings, which allow only about six expansions.

It is to be regretted that Mr. Ball does not give indicator diagrams and a fuller description of the details of the tests of his 75 horse-power compound engine. He places a brief assertion against actual indicator diagrams and modifications of their measurements based upon values of probable cylinder condensation which have been obtained by experienced engineers from numerous tests reported in detail in various publications and which stand until contradicted by other tests of accuracy. It would be of interest to compare the diagrams from his compound engine with those of the paper, and to observe the percentages of condensation, and the action of the valves.

Shaft-governed, multi-ported, four-valve high speed engines give diagrams remarkably like those from Corliss engines with releasing gear, particularly with the higher rotating speeds now used with the latter, because dashpot action is not instantaneous and shows similar rounding of diagram corners.

When steam pressures in these plants are regularly carried at 140 to 160 pounds Mr. Ball's conclusions will have a greater percentage of value than under the present usual maximum pressures of 100 to 125 pounds only, with back pressures generally specified at three pounds and often finally carried at seven to ten. I still hope for an opportunity to make a service test of engines in a city building, and so to confirm the indications that it is not so much the proportions of the compound engine or the actual valve setting as it is the moderate steam pressure and the running of engines a large part of the time under light and varying loads which reduce the gain due to compounding so that the four-valve simple engine with a fairly low water consumption is the better engine for average present conditions in city building work.

No. 1061.*

STAYBOLTS, BRACES AND FLAT SURFACES.

A DISCUSSION OF VARIOUS RULES AND FORMULÆ.

BY R. S. HALE, BOSTON, MASS.

(Member of the Society.)

Support of Flat Surfaces of Boilers by Staybolts.

1. The problem of the support of flat surfaces arises in the design of the fireboxes of locomotive and upright boilers, in the design of the combustion chambers of Scotch marine boilers and in the design of the heads above the tubes in horizontal tubular boilers of all classes.

2. The stress on the staybolts is in general a direct pull. In the case of diagonal braces the actual stress may be reduced in figures to an equivalent direct pull by considering the angle between the brace and the plate. In the case of short stays there is a bending action due to differences in expansion of the inside and outside plate which causes special stresses which will be considered below.

3. The maximum working stress permitted in staybolts is as follows:

United States Rule, 6,000 pounds per square inch.

British Board of Trade, 9,000 pounds per square inch.

British Lloyds, 8,000 pounds per square inch, if $1\frac{1}{2}$ inch diameter or less.
" " 9,000 " " " " " over $1\frac{1}{2}$ inch diameter.

German Lloyds, $\frac{1}{4}$ of tensile strength. For 60,000 pounds this is 8,560.

Bureau Veritas, $\frac{1}{5.75}$ of lower test limit, then add $\frac{1}{8}$ inch for material of 60,000 pounds: this is 10,400 on net section and about 8,300 on the gross section of a 1-inch stay.

Hartford Boiler Insurance Company, no official rule, but in the *Locomotive* of September, 1893, recommends 7,500 on long stays, and in same paper, March, 1892, recommends 4,800 on short stays.

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

None of these rules, except the practice of the Hartford Co., take any account of the difference in stress between short and long braces. Some of them allow a greater stress on large braces, as, for instance, Lloyds and Bureau Veritas.

4. Now, in short braces, such as those in locomotive fireboxes, the difference in expansion between the inside and the outside sheet bends the stays and induces severe stresses in addition to the direct pull due to the steam pressure. As a result, such staybolts are frequently found broken close to the outside plate.

5. Further, as is well known, excessive stress and strain makes the metal more liable to corrosion, and these short bolts are frequently found corroded and wasted.

6. In boilers with long braces, such as the ordinary horizontal tubular boilers, we very seldom find any corrosion. However, an allowance for corrosion would appear excellent practice, even if corrosion appear but seldom, and the rule of the Bureau Veritas, by which the stress is computed on the net area that will remain after $\frac{1}{8}$ inch ($\frac{1}{16}$ inch on each side) has been wasted, is an excellent one. For long braces where there is no bending action, say braces of 2 feet and over, there would appear to be no reason why the stress should not be as great as that allowed on similar metal in boiler plates. In boiler plates a stress of 12,000 or more is allowed between rivet holes, and from 7,000 to 11,000 in the body of the plate. It would, therefore, appear that we can safely take 12,000 for long steel braces, or 10,000 for iron. While this allows considerably more pressure than has been usual in some cases, the fact that these long braces give but little trouble would seem to warrant increasing the pressure.

The allowance of $\frac{1}{8}$ inch for future corrosion makes this agree better with current practice than is apparent at first sight. Thus, with a 1-inch brace, 12,000 pounds on the net section, which is $\frac{7}{8}$-inch diameter, is 9,200 on the gross section, or but little more than is now usual. For large braces the proposed rule will allow higher pressure than is now customary, but our experience with such cases as have come to our notice indicates that it will be amply safe.

7. On short braces the working stress due to tension alone should be less in order to leave a margin for the bending stress. These braces are usually iron, and if we make the permissible stress 9,000 on the net section after subtracting $\frac{1}{16}$ inch on all sides, on iron braces less than 24 inches long, 8,000 if less than 12 inches

long, 7,000 if less than 6 inches long, and 5,000 if less than 4 inches long, we shall get results that will be in better accord with present good practice and experience than any of the rules stated above in paragraph 3. The difference will, at least, take partial care of the stresses due to bending.

The variations above suggested according to length of braces are not based on any theory, but are chosen so that for very short braces the stress shall be about that which locomotive boilermakers have found good practice, with stresses on braces of intermediate lengths something in proportion.

In the case of the circular fireboxes of upright boilers it must be remembered that the strength of the furnace as a flue diminishes the stress on the staybolts.

8. We may now consider the strength of the plate itself, and this obviously depends on the distance between stays and the thickness of the plate.

9. It also depends somewhat on the diameter of the stay, since we can easily imagine an extreme case, in which the stays were so large as to diminish appreciably the area of plate to be supported. It is obvious, however, that for any given method of attaching the stay to the plate a change in the diameter of the stay will affect the stresses by but a very small amount; hence it will be correct to neglect this factor in determining the thickness of the plate for any stated method of attachment and distance between stays.

10. The method of attachment, of course, affects the safe working pressures.

11. The theoretical stress, independent of method of attachment, or rather for stays attached at points of infinite strength, has been found by Grashof (Unwin Machine design) to be as follows:

$$f = \text{maximum stress} = \frac{2}{9}\frac{p^2}{t^2}P.$$

p = pitch, t = thickness of plate, P = steam pressure; all in pounds and inches.

Since the stress on the bolt itself is given by the formula:

$$f = P\frac{4p^2}{\pi d^2}, \; d \text{ being the diameter of the bolt,}$$

it is interesting to note that for equal stress in bolt and plate the bolt should be 2.4 times the thickness of the plate, whatever the pitch for the case covered by the formula.

12. The stress on the plate is stated by D. K. Clark to be determined as follows: Steam pressure at elastic limit of plate equals $5{,}700 \times \frac{t}{p}$, p in this case being distance between bolts, not between centers.

Although Clark states this to be the elastic strength, yet it appears to be more probable that it is some fraction of Clark's idea of the ultimate strength. It would probably be as wrong to depend on Clark in such a case as it would be to use his theories as to strength of beams in designing bridges, instead of using the formulæ based on the elastic theory, such as the formulæ used by all the best engineers in designing every large bridge or steel frame building.

13. The rules used by the various insurance and similar bureaus are as follows:

United States, $P = C\frac{t^2}{p^2}$; $C = 112$ for steel plates $\frac{7}{16}$ or less.

British Lloyds, $P = C\frac{t^2}{p^2}$; $C = 90$ for steel plates $\frac{7}{16}$ and less.

German Lloyds, $P = C\frac{t^2}{p^2}$;

Board of Trade, $P = C\frac{(t+1)^2}{p^2 - 6}$; $C = 66$;

Bureau Veritas, $P = \frac{(t-1)^3}{2p}\frac{T}{C}$; T = tensile strength taken at 25.8 tons or 60,000 pounds in following computations.

C = .055 and less.

Hartford Boiler Insurance Co., in *Locomotive* of March, 1892, page 38, advocates D. K. Clark's rule with a factor of safety of 3, based on the elastic strength. His rule is:

$P = \frac{5{,}700\ T}{p}$, T being thickness in inches; p is the net pitch, *i.e.*, distance between staybolts and not between centers.

The figures given in the tables following take p as the distance between centers in all cases. If it was worth while to take account of the difference the figures could be corrected accordingly for Clark's formula.

The constants above given are considerably increased for other methods of attaching stays to plates than merely screwing in and riveting over.

14. In Tables I.-IV. are given the permissible pressures by these rules for various thicknesses of plates, for pitches from 4 to 11 $\frac{3}{16}$ inches.

TABLE I.

STEEL PLATES. SCREW STAYBOLTS RIVETED. PITCH, 4 INCHES.
WORKING PRESSURES.

Thickness of Plate.	British Board of Trade.	United States Rule.	Bureau Veritas.	British Lloyds.	D. K. Clark.
1/4	123 to 165	112	90	90	107
5/16	176 to 237	175	160	141	149
3/8	323	252	250	203	178
7/16	422	343	360	276	208

TABLE II.

STEEL PLATES. SCREW STAYBOLTS RIVETED. PITCH, 6 INCHES.

Thickness of Plate.	British Board of Trade.	United States Rule.	Bureau Veritas.	British Lloyds.	D. K. Clark.
1/4	55	50	40	40	79
5/16	79	78	71	62	99
3/8	108	112	111	90	120
7/16	178	214	218	177	158
1/2	266	334	360	278	198
9/16	372	480	538	400	238

TABLE III.

STEEL PLATES. SCREW STAYS RIVETED. PITCH, 8 INCHES.

Thickness of Plate.	British Board of Trade.	United States Rule.	Bureau Veritas.	British Lloyds.	D. K. Clark.
1/4	28	28	22	23	59
5/16	56	63	63	51	89
3/8	92	120	122	100	109
7/16	138	188	202	156	148
1/2	193	270	302	225	178

TABLE IV.

STEEL PLATES. SCREW STAYS RIVETED. PITCH, 11 5/16 INCHES.

Thickness of Plate.	British Board of Trade.	United States Rule.	Bureau Veritas.	British Lloyds.	D. K. Clark.
1/4	13	14	11	11	12
5/16	26	31	31	25	63
3/8	44	60	61	50	84
7/16	65	94	102	78	105
1/2	91	135	151	112	126
9/16	121	185	212	153	147

TABLE V.

RECENT CASES OF BULGES BETWEEN STAYBOLTS FOUND BY MUTUAL BOILER INSURANCE COMPANY INSPECTORS.

Working Pressure.	Pitch of Stays.	Thickness of Plate.	Pressure by U. S. Rule.
	Inches.		
65	5	$\frac{5}{16}$	112
80	$7\frac{1}{2}$	$\frac{1}{4}$	32 *
75	7	$\frac{5}{16}$	57 *
85	4	$\frac{5}{16}$	173
75	8	$\frac{1}{4}$	29 *
80	$5\frac{1}{2}$	$\frac{3}{8}$	132
100	4	$\frac{3}{8}$	250
75	4	$\frac{5}{16}$	173
70	4	$\frac{5}{16}$	173
80	8 x 7	$\frac{1}{4}$	32 *
75	4	$\frac{5}{16}$	173
60	7 x $7\frac{1}{2}$	$\frac{5}{16}$	50 *
100	$4\frac{1}{2}$ x 5	$\frac{3}{8}$	180
120	4	$\frac{3}{8}$	250 **
80	6 x $5\frac{1}{2}$	$\frac{3}{8}$	120
115	6	$\frac{3}{8}$	112 **
120	6	$\frac{3}{8}$	112 **
100	$4\frac{1}{2}$	$\frac{1}{2}$	400
60	$6\frac{1}{2}$	$\frac{3}{8}$	98
60	$4\frac{1}{2}$ x 5	$\frac{5}{16}$	123

* Small circular firebox. ** Large circular firebox.

For the cases marked * or ** the circular form of the firebox strengthens the plate against collapse, as was shown in my article in the *Engineering Magazine*.

The double figures given for the Board of Trade are the figures from their rule and the figures from Trail's Board of Trade tables for pitches under 6 inches, which tables, it is understood, supersede the rules for these small pitches.

It should, of course, be understood that probably none of the rules are intended for extreme cases. For pressure under 30 pounds various other considerations would, of course, enter and for pressures above 250 pounds temperature changes in the plate would very likely make the rules inapplicable. In other words, none of these rules are intended for extreme cases, and for ordinary working conditions it appears they give results that do not differ widely from each other.

Such being the case, the simplest formula is probably the best. Further, there should be a decided preference for formulæ that agree with the theoretic formulæ rather than for empirical formulæ. Of course, an empiric formula may represent the results of a few experiments better than a theoretic one, but unless the experiments are very numerous and very accurate the chances

are that the differences between the empiric and theoretic formulæ are due to errors of experiment.

15. In this connection it may be noted that such experiments as have been made, notably those of Fairbairn, Clark, Wilson, Greig, Eyeth and the Board of Trade (see Stromeyer, page 140, for a list of these) are not complete enough and apparently are not accurate enough to furnish any rule. This is only what is to be expected since experiments on flat plates, if the plates are tested to breaking, or even only to bulging, deal with pressures and strains far in excess of working conditions and must deal with the metal in a more or less plastic state. Experiments to determine the actual working stress in flat plates under ordinary conditions would have to be very refined experiments in which the stresses were inferred from the measurement of infinitesimal deflections, and these experiments have not been made. As stated above, the case is very similar to that of iron beams, in which the working stresses have to be computed from the elastic theory, since experiments by testing beams to destruction give results which we know are not to be relied upon in designing.

16. Hence, it follows that the theoretic form used by the United States, German and British Lloyds, etc., is the best, and that working pressure $=$ constant $\frac{t^2}{p^2}$ is the form to be adopted.

17. In regard to choice of constants it is interesting to note that on the whole the United States rule gives results intermediate between the British Board of Trade and the Bureau Veritas, although considerably higher than the British Lloyds. In order to see if these results were borne out by experience I have made up a list of the most recent cases in which our inspectors have found plates bulged between staybolts, Table V.

18. It is clear that if weakness caused these bulges, all the rules give much too high pressures, but it is more probable that these bulges are most of them due to overheating, just as the bulges in the furnaces of internally fired boilers or the bottom plates of horizontal return tubular boilers, and that they are not due to weakness of design. In other words, they are due to abnormal conditions, and are to be prevented by care in the operation of the boiler rather than by change in the design.

19. The proper constant, therefore, is somewhere between 90 and 120, and since failure will not cause serious accident it would seem wise to use 115 in determining the limiting safe pressure,

while 100 will give good results when designing a new boiler for economically long life.

The United States and Lloyds permit a slightly higher constant for thick plates than for thin, and the Bureau Veritas rule has this effect. The Board of Trade rule and Clark's have just the reverse effect. I cannot see anything either in our experience or the experiments that warrants allowing proportionate greater pressure on thick plates than called for by Grashof's formulæ, except so far as may be desired for corrosion in special cases, where it is thought corrosion should be specially provided for.

21. For braces attached by nuts or crowfoot braces attached by rivet, it is obvious that a higher pressure can be allowed. Lloyd allows about 20 per cent. more for nuts, United States about 20 per cent. more, Board of Trade about 25 per cent. more; and a constant of 140, which would agree with the margin in the other rules, is good practice.

22. For horizontal return tubular boilers the braces, when small diagonal braces are used, are attached to the heads by crowfeet and rivets. For this case a constant of 140 appears to give results in accordance with good practice.

23. For the cases where the stays or braces are in the steam space, with fire or hot gases on the other side so that the plate and ends of the braces are exposed to overheating, the British Board of Trade diminishes its constant and the allowable pressure by some 40 per cent. The other rules take no acount of such cases, but apparently allow full pressure. Such a flat surface with steam on one side could not safely be exposed to the full heat of the furnace at any pressure and yet be safe. On the other hand, if exposed to comparatively cool gases it may be as safe as if it had water on one side, instead of steam, but was exposed to the full heat of the furnace.

It follows, therefore, that no general rule can be laid down, but the constant for stays in the steam space, when the plate is exposed to the products of the combustion on the other side, may have to be reduced according to the particular case.

24. The foreign rules allow quite an excess of pressure for various washers, etc. As Stromeyer has shown, however, thicker plates are cheaper than such washers, and the use of channels or angles as in United States practice seems also to be superior. For such construction a constant of something like 200 to 250 may probably be taken, but the exact amount would depend on the

washers, the two pitches, on the shape and dimensions of the channels, on the riveting, etc., so that no rule could be laid down.

25. In the above discussion it has been assumed that the stays are all pitched in squares. If they are not, the British Board of Trade uses the square root of the surface and the Bureau Veritas the square root of the sum of the squares, for the pitch. In general, it seems to be assumed that if the pitches differ only slightly an intermediate figure as in these rules may be taken, while if they differ largely the case must be specially considered. The rule of the British Board of Trade for using the square root of surface, *i.e.*, of the product of the two pitches if they differ slightly, is simple and probably as correct as any, within the limit of say 20 per cent. variation.

SUMMARY.

Stress on Stays.

Comparison of the various rules for working stress on stays and braces shows a general neglect of the difference between long and short stays.

For long steel stays, subtracting $\frac{1}{8}$ inch from the diameter, and then allowing 12,000 pounds per square inch on net section remaining, gives results which apparently will be better than any of the present rules. For iron stays probably 10,000 should be used.

For short stays, which are chiefly iron, where bending action comes in, empiric rules reducing stress as length of stay decreases may be used. Our knowledge of the bending stress is not sufficient to warrant the use of a more theoretic formula.

Pitch of Stays and Thickness of Plates.

Comparison of the working pressures for pitch of stays and thickness of flat plates shows that the complicated formulæ of some of the rules do not give as good results as the simpler formula $P = C\frac{t^2}{p^2}$, C being taken as 100 to 115 for riveted stays, 140 for stays nutted or crowfoot stays riveted on, and a higher value up to 200 or 250 for the use of washers or channels or angle bars riveted on.

If the pitches differ by less than 20 per cent. use the surface

instead of p^2. If the pitches differ by more than 20 per cent. it is a special case. Special cases and unusual construction must always receive special consideration.

The above constants give results indicating the probable safe pressure for a year or so, until the next examination. In designing a boiler for a long life the constants should be reduced by some 20 per cent. or so, just as a factor of safety of 4 can sometimes be used temporarily for the shell of a new boiler, while 5 should be used in designing a boiler to be run for a number of years without reduction of pressure.

DISCUSSION.

Mr. Wm. T. Magruder.—As a proof of the statement made in paragraph 18 that "these bulges are due to overheating," I would refer you to an article by the writer on "The Effect of Scorching on Ductility," which will be found on page 258 of Vol. XVII of the *Stevens Institute Indicator* for July, 1900, and wherein are given the record of tests and the autographic stress-strain diagrams of coupons cut from the front, center and back of the crown-sheet of a Belpaire locomotive boiler. The locomotive was first put into service in July, 1894. In December, 1897, it was slightly scorched or overheated, due in all probability to low water. It became badly pocketed between the heads of the crow stay rivets, but there was no explosion or giving way of the sheet. The plate was purchased on specifications calling for 55,000 to 65,000 pounds per square inch of tensile strength and 28 per cent. elongation in eight inches.

TABLE.

	Front.	Center.	Back.
Size, inches	1 x 0.335	1 x 0.337	1 x 0.334
Area, before testing, sq. in.	0.335	0.337	0.334
Area, after testing, sq. in.	0.240	0.271	0.144
Per cent. reduction in area	28.4	19.6	56.9
Strength, lb. per sq. in., at elastic limit (yield point)	38,060.	37,090.	37,040.
" " " " maximum	60,300.	56,380.	63,270.
" " " " final	53,730.	48,960.	52,470.
Elongation in eight inches	1.09	0.53	1.97
Per cent. of elongation	13.6	6.6	24.6

It is to be noted that the 28 per cent. elongation, when new, in a test section eight inches long was reduced to 24.6 per cent. at the back coupon, to 6.6 per cent. at the center, and to 13.6 per cent. at the front coupon, after the use that it had received; that

the reduction in area was decreased from 56.9 per cent. at the back to 19.6 per cent. at the center and to 28.4 per cent. at the front test-section; and that the maximum strength was reduced from 63,270 pounds per square inch at the back to 56,380 pounds per square inch at the center. The center coupon showed the highest modulus of elasticity. It gave a rounded diagram, whereas the back coupon gave a rather sharp corner at the yield point.

The center coupon, after fracture, showed a ruptured or checked surface on the side of the plate which was next to the water, and in places originally about four inches apart and midway between the crown-stay rivets. It was quite uniformly but less deeply checked on the fire side. The front coupon was checked only on the fire side, and only near the place of fracture, which was at a line of rivets.

While the sheet did not give way and cause a steam boiler explosion, with the attendant loss of life and property, the tests show that the ductility of the sheet had been practically destroyed, and confirm the judgment of the person who ordered it to be replaced.

The checking of these plates between the crown-stay rivets would seem to indicate that the heat is not conducted away as rapidly between as at the crown-stay rivets, and that therefore they become scorched, the rivets acting like fins. This result may be due either to the poor circulation of the water over the crown sheet or to the existence of the water in the spheroidal state, or to both.

Mr. G. R. Henderson.—Paragraph 18 on page 183 is, I think, particularly interesting.

The last part of the paragraph, after referring to bulges between stay bolts, states:

". . . It is more probable that these bulges are most of them due to overheating, just as the bulges in the furnaces of internally fired boilers, or the bottom plates of horizontal return tubular boilers, and that they are not due to weakness of design. In other words, they are due to abnormal conditions, and are to be prevented by care in the operation of the boiler rather than by change in the design."

I would like to cite some cases in connection with locomotive boilers that came under my own experience. The inside fire-box sheets were $\frac{3}{8}$ thick, the spacing was four inches, and the pressure to be withstood 200 pounds. According to the United

States rules the safe pressure would be 250 pounds, so you see it was quite within the safe limits, and it was not a question of pressure. We found the side sheets bulged very badly between the stay bolts.

Examination showed that there were many minute cracks started apparently on the water side, and they kept getting worse and worse until they came clear through the sheets. There seemed to be no question but what there was plenty of water. The boiler was of the wide fire box type. It occurred to some of us that possibly the generation of steam was so rapid on account of the large mass of incandescent fuel that the water next to the inside of the box was converted into steam.

In order to make a test of this we introduced a series of gauge cocks from the outside of the shell reaching in near the fire box sheet. One stopped within a quarter of an inch and the other within half an inch and the other about three-quarters of an inch. We found that when the engine was working hard the cock close to the fire box gave indications of steam without water. When the throttle valve was closed water was obtained at all the gauge cocks. But when the engine was working hard the gauge cock nearest the sheet showed steam generally. The test seemed to indicate that there was a $\frac{3}{4}$-inch film of steam next to the fire box. Now according to this statement on page 183, that could have been remedied by care in the operation of the engine. But I do not see how that could have been done in this case, because in the operation of a locomotive we are bound to get all we can out of it. I think if any one can give us information in regard to the film of steam being next to the fire box it might be instructive. If that is the case, as we found it was in this experiment, I don't believe if we had a water leg 12 inches wide that we would have any better result. This one was 4 inches wide at the bottom and 7 inches at the crown sheet and it seemed from the experiment that the radiant heat was so intense that the water was apparently driven from the sheet just the same as when you throw a cupful of water on a red hot stove.

Mr. W. S. Huson.—Some members will remember a small section of boiler plate showing a bulge, and exhibited in connection with a paper read before the Society, say ten years ago.* In the discussion, the question of water being driven away from the

* *Transactions* A. S. M. E., vol. xiv, No. 511.

shell was raised, and as Mr. Henderson has referred to this, if the paper is traced back, a very full discussion will be found.

*Mr. Robt. S. Hale.**—Mr. Henderson's criticism is correct. It is the design of the staybolting only that I referred to. Undoubtedly a change in the design so as to keep the surfaces covered by water would prevent many of these bulges.

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* Author's Closure, under the Rules.

No. 1062.*

LOSSES IN NON-CONDENSING ENGINES.

BY JAMES B. STANWOOD, CINCINNATI, O.

(Member of the Society.)

1. Some years ago, the writer became interested in the subject of the possibility of increasing the economical performance of non-condensing steam engines, operating under varying loads. At that time through the kindness of Professor Sweet, he was fortunate enough to become acquainted with the work of Mr. Willans, as set forth in his two celebrated papers upon "Non-condensing and Condensing Engine Trials." † Mr. Willans applied in a most practical way the extremely theoretical and technical laws of thermodynamics, simplified, however, by the use of the Temperature-Entropy Diagram.

2. After the first tentative report of the A. S. M. E. Committee on Standardizing Engine Tests, the writer contributed to the discussion of the report by testing some of the recommendations made by the Committee by Mr. Willans' methods of analysis; calling special attention to two factors, $1-C$ and $1-F$ (suggested by the Committee), respectively representing the condensation and expansion losses. This paper, therefore, is an extension or elaboration of the discussion upon the relation of these losses, especially such as occur in non-condensing engines. The tests of a few leading types of these engines which have been analyzed in order to determine current practice, and some of the conditions governing steam economy under varying loads are likewise given.

3. Mr. Willans, in recording the analyses of tests made upon his special type of engine, compares in each test the number of British

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† The Proceedings of the Institute of Civil Engineers of Great Britain, vol. xciii, 1888 and vol. cxiv, 1893. The writer is also greatly indebted to Mr. Willans' partner, Mr. Mark Robinson, for copies of these papers and for other valuable information.

thermal units actually converted into useful work per pound of H_2O,* with the theoretical number of British thermal units per pound of steam which theory dictates as available.† The ratio expressing this comparison of practice with theory is an efficiency, called by Professor Reeves the "cylinder efficiency," a term which we shall use and denote algebraically by the letter E.

4. In calculating the theoretically available British thermal units, Mr. Willans employed the Rankine cycle as being the one that gives the highest practically obtainable efficiency.‡ In this cycle, steam is assumed to expand adiabatically, instead of by either of two more common assumptions: one in which the expansion curve follows the law, pv=constant, and one in which it follows the curve of perfectly saturated steam.

5. The relative efficiency of these three methods is analyzed and discussed by Mr. Willans, who gives his reasons for the selection of this special cycle. Capt. Henry Riall Sankey later discussed this same question more elaborately in a paper presented to the Institute of Civil Engineers in 1895-96 entitled "The Thermal Efficiency of Steam Engines."

6. The usual computations, by means of Steam and Hyperbolic Logarithmic Tables, to determine theoretical values, in British thermal units, in accordance with the Rankine cycle are laborious. It is simpler to use a table of Absolute Pressures and Temperatures in connection with their respective Entropy values of water and steam. Such a table is Table I., calculated by the writer from Peabody's Steam Tables.¶ It can be used as follows:

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* H_2O represents mixture steam and water.

† This theoretically available number of British thermal units is the difference between the total heat per pound of steam delivered to an engine, and the theoretical heat per pound of H_2O rejected by that engine at the given terminal and back pressures.

‡ Since his time the Nordberg system of progressive feed-water heating, has brought into prominence the idea of making the more efficient Carnot cycle approximately a mechanical possibility.

¶ There is considerable variation in the values given in Entropy Tables as computed by different authorities, due to variations among the steam tables used as basis of calculation. This table, however, closely approximates Professor Reeves' Entropy Tables which have recently been published; his notation is used in this paper so that those who will, may more easily use his tables. He takes absolute zero at 460.8 below Fahrenheit zero; here 461 degrees below is taken. For some reasons Table I might have been omitted from this paper but for the fact that it contains values of the weights of 5.4 cubic feet of steam—a quantity needed in calculations, especially when associated with entropy values.

TABLE I.

Absolute Pressures, Temperatures, Entropy and Weights of 5.4 Cubic Feet of Steam.

Absolute Pressure P, in lb.	θ Absolute Temperature T, Fahrenheit.	Entropy = ϕ. of Water Nw in Ranks.	Entropy = ϕ. of Steam Ns in Ranks.	Weight of 5.4 cu. ft. of Steam W, in lb.
0	493.0	0.000	2.214	0.0016
0.5	540.8	0.093	2.049	0.0083
1	563.0	0.133	1.986	0.0162
2	587.3	0.175	1.923	0.0311
3	602.6	0.201	1.886	0.0456
4	614.1	0.220	1.860	0.0598
5	623.3	0.235	1.841	0.0738
6	631.1	0.248	1.825	0.0876
7	637.9	0.259	1.811	0.1012
8	643.9	0.268	1.800	0.1148
9	649.3	0.277	1.790	0.1282
10	654.3	0.284	1.781	0.142
11	658.8	0.291	1.772	0.155
12	663.0	0.298	1.765	0.168
13	666.9	0.304	1.758	0.181
14	670.6	0.309	1.752	0.194
14.7	673.0	0.313	1.748	0.203
15	674.0	0.314	1.746	0.207
16	677.3	0.319	1.741	0.220
17	680.4	0.324	1.736	0.233
18	683.4	0.328	1.731	0.246
19	686.2	0.332	1.726	0.258
20	689.0	0.336	1.722	0.271
21	691.6	0.340	1.718	0.284
22	694.1	0.344	1.714	0.297
23	696.5	0.347	1.710	0.310
24	698.8	0.351	1.707	0.322
25	701.0	0.354	1.703	0.335
26	703.2	0.357	1.700	0.347
27	705.3	0.360	1.697	0.360
28	707.4	0.363	1.694	0.373
29	709.3	0.366	1.691	0.385
30	711.3	0.369	1.684	0.397
31	713.2	0.371	1.685	0.410
32	715.0	0.374	1.683	0.422
33	716.8	0.376	1.681	0.435
34	718.5	0.379	1.678	0.447
35	720.2	0.381	1.676	0.459
36	721.9	0.383	1.674	0.472
37	723.5	0.386	1.672	0.484
38	725.1	0.388	1.669	0.496
39	726.6	0.390	1.667	0.509
40	728.1	0.392	1.665	0.521
41	729.6	0.394	1.663	0.533
42	731.1	0.396	1.661	0.545
43	732.5	0.398	1.660	0.557
44	733.9	0.400	1.658	0.569
45	735.3	0.402	1.656	0.582
46	736.7	0.404	1.654	0.594
47	738.0	0.406	1.652	0.606
48	739.3	0.407	1.651	0.618
49	740.6	0.409	1.649	0.630
50	741.9	0.411	1.648	0.642
51	743.1	0.413	1.646	0.654
52	744.3	0.414	1.645	0.666
53	745.5	0.416	1.643	0.678
54	746.7	0.418	1.642	0.690
55	747.9	0.419	1.641	0.701
56	749.1	0.421	1.640	0.713
57	750.2	0.422	1.638	0.726
58	751.3	0.424	1.636	0.738
59	752.4	0.425	1.635	0.749
60	753.5	0.427	1.633	0.761
61	754.6	0.428	1.632	0.773
62	755.7	0.430	1.631	0.785
63	756.7	0.431	1.629	0.797
64	757.7	0.432	1.628	0.808
65	758.8	0.434	1.627	0.820
66	759.8	0.435	1.626	0.832
67	760.8	0.436	1.625	0.843
68	761.8	0.438	1.623	0.855
69	762.7	0.439	1.622	0.867
70	763.7	0.440	.....	0.879
71	764.7	0.442	1.620	0.890
72	765.6	.....	.....	0.902
73	766.5	0.444	1.618	0.914
74	767.5	.....	.....	0.926
75	768.4	0.446	1.616	0.937
76	769.3	0.448	1.615	0.949
77	770.2	.....	.....	0.961
78	771.1	0.450	1.613	0.973
79	772.0	.....	.....	0.984
80	772.8	0.452	1.611	0.995
81	773.7	0.453	.....	1.007
82	774.5	0.455	1.609	1.018
83	775.4	.....	.....	1.030
84	776.2	0.457	1.607	1.042
85	777.0	.....	.....	1.054
86	777.8	0.459	1.606	1.065
87	778.7	.....	.....	1.077
88	779.5	0.461	1.604	1.089
89	780.3	.....	.....	1.100
90	781.0	0.463	1.602	1.111
91	781.8	.....	.....	1.123
92	782.6	0.465	1.601	1.135
93	783.4	.....	.....	1.146
94	784.1	0.467	1.599	1.158
95	784.9	.....	.....	1.169
96	785.6	0.469	1.597	1.180
97	786.4	.....	.....	1.192
98	787.1	0.471	1.596	1.204
99	787.9	.....	.....	1.215
100	788.6	0.473	1.594	1.226
101	789.3	.....	.....	1.238
102	790.0	0.475	1.593	1.250
103	790.7	.....	.....	1.261
104	791.4	0.477	1.591	1.272
105	792.1	.....	.....	1.284
106	792.8	0.479	1.590	1.295
107	793.5	.....	.....	1.307
108	794.2	0.481	1.589	1.318
109	794.9	.....	.....	1.329
110	795.6	0.483	1.587	1.341
111	796.2	.....	.....	1.353
112	796.9	0.484	1.586	1.364
113	797.6	.....	.....	1.375
114	798.2	0.486	1.584	1.387

TABLE I.—(*Continued*).

Absolute Pressure P, in lb.	θ Absolute Temperature T, Fahrenheit.	Entropy = ϕ. of Water N_W in Ranks.	Entropy = ϕ. of Steam N_S in Ranks.	Weight of 5.4 cu. ft. of Steam W, in lb.	Absolute Pressure P, in lb.	θ Absolute Temperature T, Fahrenheit.	Entropy = ϕ. of Water N_W in Ranks.	Entropy = ϕ. of Steam N_S in Ranks.	Weight of 5.4 cu. ft. of Steam W, in lb.
115	798.9	.....	.....	1.398	175	831.7	0.529	1.553	2.074
116	799.5	0.488	1.583	1.409	176	832.1	.....	.....	2.085
117	800.1	.....	.....	1.421	177	832.6	0.530	.....	2.097
118	800.8	0.489	1.582	1.433	178	833.1	0.531	1.552	2.108
119	801.4	.....	.....	1.444	179	833.5	.....	.....	2.119
120	802.1	0.491	1.581	1.455	180	834.0	0.532	.....	2.130
121	802.7	.....	.....	1.466	181	834.4	.....	1.551	2.142
122	803.3	0.493	1.579	1.477	182	834.9	0.533	.....	2.153
123	803.9	.....	.....	1.489	183	835.3	.....	.....	2.164
124	804.5	0.494	1.578	1.501	184	835.8	0.534	1.550	2.175
125	805.1	.....	.....	1.512	185	836.2	0.535	.....	2.186
126	805.7	0.496	1.577	1.523	186	836.7	.....	1.549	2.198
127	806.3	0.497	.....	1.534	187	837.1	0.536	.....	2.209
128	806.9	.....	1.576	1.545	188	837.6	.....	.....	2.220
129	807.5	0.498	.....	1.557	189	838.0	0.537	1.548	2.231
130	808.1	.....	1.575	1.568	190	838.4	.....	.....	2.243
131	808.7	0.500	.....	1.580	191	838.9	0.538	.....	2.254
132	809.3	.....	1.574	1.591	192	839.3	.....	1.547	2.265
133	809.9	0.501	.....	1.602	193	839.8	0.539	.....	2.276
134	810.5	.....	1.573	1.614	194	840.2	.....	1.546	2.287
135	811.0	0.503	.....	1.625	195	840.6	0.540	.....	2.299
136	811.6	.....	1.572	1.636	196	841.0	.....	.....	2.310
137	812.2	0.504	.....	1.648	197	841.5	0.541	1.545	2.321
138	812.7	0.505	1.570	1.659	198	841.9	0.542	.....	2.332
139	813.3	.....	.....	1.670	199	842.3	.....	.....	2.343
140	813.9	0.506	1.569	1.681	200	842.7	0.543	1.544	2.354
141	814.4	.....	.....	1.692	201	843.2	.....	.....	2.365
142	815.0	0.508	1.568	1.704	202	843.6	0.544	.....	2.376
143	815.6	.....	.....	1.715	203	844.0	.....	1.543	2.387
144	816.1	0.509	1.567	1.726	204	844.4	0.545	.....	2.398
145	816.6	0.510	.....	1.738	205	844.8	.....	.....	2.409
146	817.1	.....	1.566	1.749	206	845.2	0.546	1.542	2.420
147	817.7	0.511	.....	1.760	207	845.6	.....	.....	2.432
148	818.2	0.512	1.565	1.771	208	846.1	0.547	.....	2.443
149	818.7	.....	.....	1.782	209	846.5	.....	1.541	2.454
150	819.3	0.513	1.564	1.793	210	846.9	0.548	.....	2.465
151	819.8	0.514	.....	1.805	211	847.3	.....	.....	2.476
152	820.3	.....	1.563	1.816	212	847.7	0.549	1.540	2.488
153	820.8	0.515	.....	1.827	213	848.1	.....	.....	2.499
154	821.3	0.516	.....	1.839	214	848.5	0.550	.....	2.510
155	821.9	.....	1.562	1.850	215	848.9	.....	1.539	2.521
156	822.4	0.517	.....	1.861	216	849.3	0.551	.....	2.533
157	822.9	0.518	1.561	1.872	217	849.7	.....	.....	2.544
158	823.4	.....	.....	1.884	218	850.2	0.552	1.538	2.555
159	823.9	0.519	1.560	1.895	219	850.6	.....	.....	2.566
160	824.4	.....	.....	1.906	220	850.9	0.553	.....	2.577
161	824.9	0.520	1.559	1.918	221	851.2	.....	1.537	2.588
162	825.4	.....	.....	1.929	222	851.6	0.554	.....	2.599
163	825.9	0.522	.....	1.940	223	852.0	.....	.....	2.610
164	826.4	.....	1.558	1.952	224	852.4	.....	.....	2.622
165	826.9	0.523	.....	1.963	225	852.8	0.555	1.536	2.633
166	827.4	.....	1.557	1.974	230	854.7	0.558	1.534	2.689
167	827.9	0.524	.....	1.985	235	856.6	0.560	1.533	2.744
168	828.3	0.525	1.556	1.996	240	858.4	0.562	.....	2.800
169	828.8	.....	.....	2.007	245	860.2	0.564	1.530	2.856
170	829.3	0.526	.....	2.018	250	862.0	0.567	.....	2.912
171	829.8	.....	1.555	2.029	255	863.7	0.569	.....	2.968
172	830.2	0.527	.....	2.040	260	865.5	0.571	.....	3.025
173	830.7	0.528	1.554	2.051	265	867.2	0.573	1.525	3.081
174	831.2	.....	.....	2.063					

7. The number of British thermal units, U, available for work in one pound of steam expanding adiabatically from an absolute pressure, p (corresponding absolute temperature being T), to a

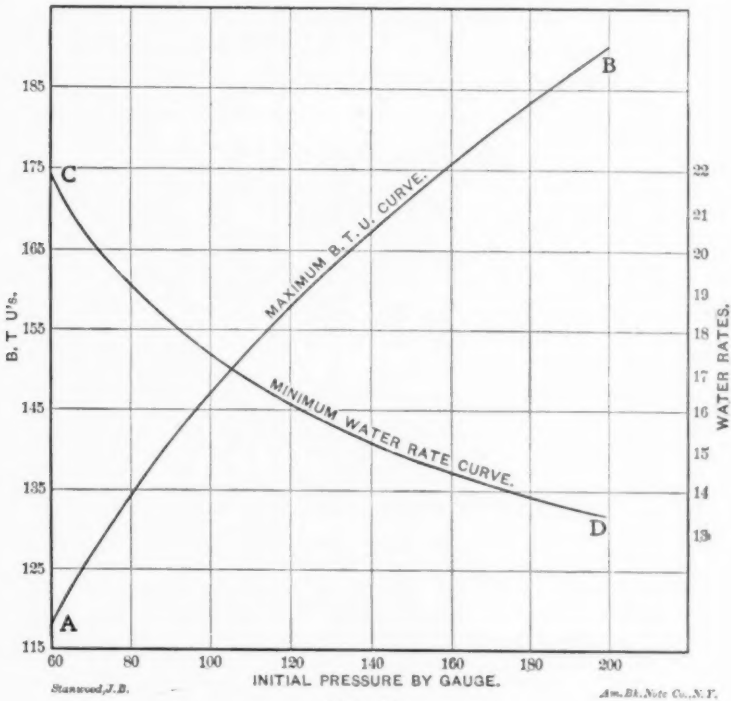

FIG. 51.

terminal pressure p_1 (corresponding absolute temperature being T_1), with a back pressure equal to terminal pressure, p_1, is:

$$*U = D\,(1 + N_s - N_w) - T_1\,(N_w - N_{w1}) \quad . \quad . \quad . \quad (1)$$

here $D = T - T_1$,

N_s = Entropy or φ of steam at T,

N_w = Entropy or φ of water at T,

N_{w1} = Entropy or φ of water at T_1.

U_f, or the mechanical equivalent of U expressed in foot-pounds = 778 U; and S, or the corresponding minimum theoretical water rate $= \dfrac{2{,}545}{U}$. The mean effective pressure corresponding to differ-

* See Appendix C.

ent boiler pressures (steam expanding adiabatically to a terminal pressure equal to atmospheric pressure) can also be determined; the expression of its value being $M = \frac{5.4\ w_1 U}{x} = \frac{W_1 U}{x}$ where W_1 =weight of 5.4 cubic feet of steam at atmospheric pressure, x= dryness factor after adiabatic expansion.*

8. Fig. 51, by means of two curves, AB and CD, gives values of the available British thermal units, U, and the minimum theoretical water rates, S, for gauge pressures ranging from 60 to 200 pounds. Table II gives values for U, S and M for the same range of gauge pressures. In the calculation of all of these values the absolute temperature of both the terminal and back pressure has been taken at 673 degrees F., corresponding to 212 degrees above the Fahrenheit zero.

TABLE II.

Boiler Pressure by Gauge.	Available B.T.U.	Minimum Theoretical Water Rate.	Corresponding Theoretical M.E.P.
P.	U.	S.	M.
60	117.0	21.8	26.1
70	126.3	20.2	28.3
80	133.6	19.1	30.3
90	141.0	18.1	32.0
100	147.3	17.3	33.6
110	152.3	16.7	35.1
120	157.6	16.2	36.5
130	162.6	15.7	37.8
140	167.4	15.3	39.0
150	171.7	14.8	40.1
160	175.6	14.5	41.2
170	179.3	14.2	42.3
180	183.2	13.9	43.3
190	186.8	13.6	44.2
200	190.2	13.4	45.2

Cylinder Efficiency.

9. The cylinder efficiency, E, of an engine working between given limits of pressure may be determined in three ways: First, by ascertaining the ratio of the British thermal units or U_y converted into work per pound of H_2O, to the corresponding avail-

* The derivation of these expressions can be found in Appendix *G*.

able British thermal units or U, $\therefore E = \frac{U_y}{U}$; or Second, by ascertaining the ratio of the actual water-rate,* S_y, to the minimum theoretical water-rate, S, $\therefore E = \frac{S_y}{S}$; or Third, by ascertaining the ratio of the actual number of foot-pounds developed per pound of H_2O, U_{fy}, to the theoretical foot-pounds, U_f, $\therefore E = \frac{U_{fy}}{U_f}$.

10. Of these the second ratio is most readily determined, as S_y is a most important determination of an engine test, and S can be found on Fig. 51, or Table II.

11. "Engine Tests," a compilation by Mr. Geo. H. Barrus, a member of our Society, gives the result of a large number of tests upon steam engines which he has conducted. These afford current data from which values of E for well-known popular types of non-condensing engines can be ascertained. Table III gives some values based on Mr. Barrus' tests; it also gives values derived from a test made on Mr. Willan's engine and from a test made by Mr. E. G. Armstrong on a single-valve automatic engine.

Losses.

12. It will be noted that in none of these tests, except that on Mr. Willans' engine, does the cylinder efficiency, E, very much exceed 0.70; in most cases it is less. The losses which cause this lack in efficiency may be grouped in one or two classes.

13. First, those losses which in the aggregate cause a deficiency between the total steam supplied to an engine and that accounted for by the indicator diagram at cut-off (or in compound engines, at high pressure cut-off). These for brevity may be called the Condensation losses.

14. Second, those which are caused by the imperfect expansion of the steam which is accounted for at high pressure cut-off by the indicator, so that the work done is less than the theoretical work available from this steam, if dry. These may be called the Expansion losses.

* A correction of the actual water-rate must be made when superheated steam is used, or when steam or water or both from jackets and receivers are returned to boilers. Steam supplied to jackets and reheaters should always be considered in these calculations as part of steam delivered to and used by an engine.

TABLE III.

CYLINDER EFFICIENCIES, CONDENSATION AND EXPANSION LOSSES IN NON-CONDENSING STEAM ENGINES.

By Whom Tested.	TYPE.	Size.	R. P. M.	M. E. P.	Water Rate.	Steam Pipe Pressure by Gauge.	Cut-off.	Cylinder Efficiency *E*.	Condensation Factor *C*.	Condensation Loss 1-*C*.	Expansion Factor *F*.	Expansion Loss 1-*F*.
		ins.				*lb.*						
Barrus, No. 23..	Single Valve, Automatic.....	8 x 12.....	303.7	24.5	31.78	83	.203	.59	.642	.358	.92	.08
			307.8	34.8	31.2	82.4	.312	.696	.72	.28	.84	.16
			304.5	43.1	30.63	81.9	.378	.611	.762	.238	.80	.20
Barrus, No. 1...	Corliss, Simple.....	23 x 60	74.7	33.1	27.77	72.8	.367	.718	.84	16	.85	.15
Barrus, No. 2..	Corliss, Simple.....	28½ x 59½.....	64.8	41.2	25.8	101	.315	.671	.817	.183	.82	.18
Barrus, No. 31..	Corliss, Twin.....	2-16 x 42.....	87.6	5.	73.6	101.8	.041	.230	.382	.618	.62	.38
			....	*13.3	38.4	.....	.084	.453	.509	.491	.89	.11
			....	*19.4	31.4	.....	.121	.552	.588	.412	.94	.06
			....	*20.5	25.8	.....	.178	.673	.709	.291	.945	.055
	* M. e. p's based on 85 r. p. m. *E* based on 100 lbs. pressure.		....	*38.1	25.4	.....	.231	.685	.745	.255	.92	.08
			84.9	48.4	25.91	98.6	.323	.670	.796	.204	.84	.16
E. J. Armstrong	Single Valve, Automatic.....	13 x 12.....	282	3.5	52.8	100	.052	.333	.300	.672	1.11	.11
			281	19.	31.1	.....	.130	.559	.62	.38	.901	.099
			281	32.1	29.2	.....	.210	.582	.66	.34	.88	.12
			280	45.5	28.75	.....	.320	.605	.72	.28	.84	.16
			279	56.5	28.8	.....	.420	.604	.775	.225	.78	.22
	E based on 100 lb. pressure.		277	67.6	30.25	.....	.530	.575	.79	.21	.73	.27
Barrus, No. 20b.	4-Valve, Gridiron Type.	28 x 60.....	50.5	23.8	30.16	65.1	.222	.67	.722	.278	.92	.08
Barrus, No. 54..	Single Valve, Compound.....	12 x 20 x 13...	275.7	18.3	24.99	166.9	.224	.572	.686	.314	.834	.166
			271.2	33.8	21.14	166.8	.379	.676	.744	.256	.91	.09
			273.4	43.2	21.25	164.6	.520	.676	.814	.186	.83	.17
Barrus, No. 44..	Vertical Single Acting Compound.....	11 x 19 x 11...	296.3	46.1	22.91	135.9	.605	.684	.803	.197	.85	.15
Barrus, No. 45e.	2-Valve, Riding Cut-off Compound.....	14 x 28 x 24...	165 6	21.8	23.24	118	.487	.706	.794	.216	.89	.11
Barrus, No. 46..	4-Valve Compound..... { A / B }	17½ x 28 x 48..	99	22.15	22.11	128.7	.309	.711	.806	.194	.88	.12
				34.4	21.59	135.5	.419	.712	.816	.184	.87	.13
P. W. Willans..	Willans' Vertical Compound.....	$9\frac{7}{16}$ x $13\frac{7}{16}$ x 6.	405	41.85	20.35	135.4	.392	.758	.883	.117	.86	.14

15. The relation of these losses to one another and to the cylinder efficiency, E, can be best illustrated graphically. In Fig. 52, $MPRS$ represents an ideal indicator diagram, Rankine cycle, of an engine whose cylinder is of such capacity that at each stroke *one pound* of steam is used to best theoretical advantage. The

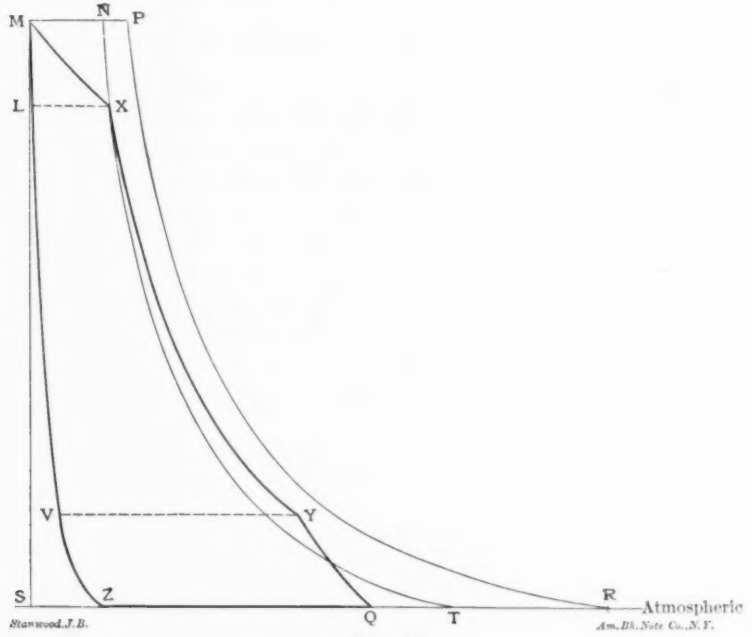

FIG. 52.

area in British thermal units, or U, see formula (1) and its water-rate, S, can be obtained from Fig. 51; the steam pipe pressure being MS, MP represents the volume of this one pound of steam at this pressure at cut-off; the terminal equal to the back pressure is identical with the atmospheric pressure.

Suppose, of the volume of steam MP delivered, that MN be the portion brought into evidence by the indicator at high pressure cut-off. Then NP represents the portion of each pound supplied corresponding to the condensation loss. Let $C = \frac{MN}{MP}$, represent the quality of steam at high pressure cut-off or the ratio of dry steam to total H_2O present at that point; then $1 - C = \frac{NP}{MP}$ is the

ratio of condensation to steam supplied, and expresses the condensation losses.

In the same diagram suppose the indicator card $MXYQZ$ to have been taken from an actual engine using a total of one pound of H_2O per stroke. The ratio of its area to the area of $MPRS$ will be its cylinder efficiency, E. If in this engine the part of the one pound of H_2O delivered to the cylinder, condensed in the form of water and represented by NP, be entirely eliminated from the cylinder at each stroke so that steam remaining is always initially dry, then this portion MN (equal in weight to LX at actual cut-off on card) would give by complete adiabatic expansion the diagram $MNTS$. The ratio $\frac{MXYQZ}{MNTS}$, or the actual to theoretical work, can be represented by a factor, F, which may be called the factor of complete expansion; and $1-F$ will be the ratio of work, lost by incomplete expansion, to the work which is due to the perfect expansion of the steam initially dry at high pressure cut-off, expresses the expansion losses.

16. Upon consideration it will be seen that

$$E = CF \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

so that, if any two of the quantities E, C or F are known the third can be obtained. E and C can only be secured by a carefully conducted test of a steam-engine; the value of F can be calculated from measurements that may be taken from any indicator card, for*

$$F = \frac{M \cdot r}{W b U} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

U may be found on Fig. 51, or in Table II, or may be calculated by (1).

17. In Table III may be found values of C, $1-C$, F and $1-F$ for the engines for which E has already been determined.

18. When $F = 1$ (all of the expansion losses eliminated) the actual ratio of expansion R equals the ratio between the volume of one pound of dry steam at initial pressure and its volume after complete adiabatic expansion to back or atmospheric pressure. To determine the volume of this steam it is necessary to ascertain the "quality," x, at terminal pressure, for, as is well known, a portion is condensed (according to theory) when doing work adiabatically.

* See Appendix *F*.

19. This quality is expressed by the factor x* indicating the shrinkage in volume as described. In appendices A and B will be found a method of determining its values, and it may be calculated by the entropy of steam and water, Table I, as follows:

$$x = \frac{Ns - Nw_1}{Ns_1 - Nw_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Ns_1=entropy of steam at terminal pressure.

In Table IV may be found values of x and R for different initial and terminal pressures.

The Expansion Factor "F."

20. When expansion in non-condensing engines is such that terminal pressure does not reach atmospheric pressure, there is a "terminal drop" in pressure and F is reduced by an amount due to loss by free expansion of steam; or if expansion is so great as to give a terminal less than atmospheric pressure, then there will be a "loop" and a loss in F due to work in compressing the atmosphere. Fig. 59 Appendix A indicates in an exaggerated way the relation between the saturation and adiabatic curves; it also shows the losses due to a terminal drop and to a loop.

21. In table IV will also be found values of F for expansions carried to different terminal from different initial pressures, steam assumed to be initially dry, expanding adiabatically.

22. There are other losses which reduce the value of F; among such are: free expansion in clearance spaces, heavy initial condensation reducing initial pressure at early cut-offs, wire drawing through small port openings, radiation, leakage through exhaust valves and past piston—these can only be determined by experience with separate engines or types of engines.

23. On the other hand there are some conditions tending to increase F, but at the corresponding expense of C. A leaky steam admission valve will continue to pass steam after cut-off and thereby increase the apparent work done by expansion. Re-evaporation of steam originally condensed, or condensation of steam during the compression period will both add to the area of

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* See "Report of Committee on Standardizing Engine Tests," American Society of Mechanical Engineers *Transactions*, vol. xxiv, page 757; also Willans' "Condensing Trials," Appendix II, in Proceedings of the Institution of Civil Engineers of Great Britain, vol. cxiv, 1893

TABLE IV.

Values for Adiabatic Expansion, Water-rates, Mean Effective Pressure, etc.

Absolute Terminal Pressure.	Quality at Terminal Pressure	Ratio of Expansion.	Factor of Expansion.	Theoretical Water Rate.	Theoretical M.E.P.	Theoretical B.T.U. per lb. steam.
p_1.	x.	R.	F.	S.	M.	U_1.
	60 lb. Initial Pressure, by Gauge.					
40	.96	1.72	.81	27.0	52.4	94.3
30	.94	2.2	.90	24.3	44.3	105.0
20	.92	3.18	.98	22.3	33.5	114.1
14.7	*.91*	*4.18*	*1.00*	*21.8*	*26.1*	*117.0*
10	.89	5.86	.96	22.8	17.8	111.7
8	.88	7.15	.89	24.5	13.6	103.9
6	.87	9.23	.74	29.4	8.9	86.7
4	.85	12.83	.39	57.9	3.1	44.1
	80 lb. Initial Pressure, by Gauge.					
60	.97	1.5	.70	27.4	73.0	92.8
50	.96	1.75	.76	25.0	67.8	101.7
40	.95	2.1	.83	23.0	61.4	110.5
30	.93	2.7	.91	21.1	52.4	120.6
20	.91	3.9	.98	19.5	39.5	130.2
14.7	*.90*	*5.1*	*1.00*	*19.1*	*30.3*	*133.6*
10	.88	7.2	.96	20.0	20.8	128.1
8	.87	8.7	.90	21.2	15.9	120.3
6	.86	11.3	.77	24.6	10.6	103.5
4	.84	16.3	.46	41.6	4.4	61.1
	100 lb. Initial Pressure, by Gauge.					
60	.96	1.94	.72	24.0	83.7	105.5
50	.95	2.06	.73	22.0	78.2	115.8
40	.94	2.51	.85	20.3	69.6	125.2
30	.92	3.23	.92	18.8	58.4	135.7
20	.90	4.62	.98	17.6	43.5	144.4
14.7	*.89*	*6.07*	*1.00*	*17.3*	*33.6*	*147.3*
10	.87	8.53	.95	18.2	22.9	140.1
8	.86	10.42	.91	19.0	18.0	134.3
6	.84	13.33	.81	21.5	12.4	118.3
4	.83	19.33	.51	33.7	5.4	75.5
	125 lb. Initial Pressure, by Gauge.					
70	.96	1.82	.71	22.5	103.7	113.1
60	.95	2.08	.75	21.1	97.3	120.7
50	.94	2.45	.81	19.8	88.4	129.1
40	.92	2.97	.86	18.4	78.2	138.4
30	.91	3.84	.93	17.1	65.0	148.9
20	.89	5.50	.98	16.2	48.0	157.6
14.7	*.88*	*7.23*	*1.00*	*15.9*	*37.2*	*160.3*
10	.86	10.13	.97	16.4	25.7	155.1
8	.85	12.4	.92	17.2	20.0	147.3
6	.84	16.1	.82	19.5	13.8	131.1
4	.82	23.0	.56	28.4	6.5	89 6
3	.81	29.7	.28	57.1	2.5	44.6
	150 lb. Initial Pressure, by Gauge.					
70	.95	2.11	.73	20.4	115.5	124.7
60	.94	2.41	.77	19.2	107.7	132.5
50	.93	2.82	.82	18.1	97.6	140.5
40	.92	3.43	.87	17.0	85.6	149.8
30	.90	4.45	.94	15.9	70.6	160.2
20	.88	6.33	.98	15.1	51.8	168.4
14.7	*.87*	*8.38*	*1.00*	*14.8*	*40.1*	*171.7*
10	.85	11.75	.97	15 3	27.7	166.4
8	.84	14.3	.93	16.0	21.7	159.0
6	.83	18.5	.83	17 9	15.1	142.5
4	.82	26.6	.59	25.1	7.4	101.2
3	.81	34.3	34	44.4	3.3	57.4
	175 lb. Initial Pressure, by Gauge.					
70	.94	2.39	.74	18.8	126.0	135.1
60	.93	2.73	.79	17.8	117.0	142.9
50	.92	3.20	.83	16.9	105.5	150.9
40	.91	3.90	.88	15.9	91.9	160.0
30	.89	5.04	.94	14.9	75.7	170.3
20	.88	7.23	.98	14.2	55.3	178.6
14.7	*.86*	*9.50*	*1.00*	*14.0*	*42.9*	*181.6*
10	.84	13.3	.97	14.4	29.7	176.6
8	.83	16.3	.93	15.1	23 3	169.1
6	.82	21.0	.84	16.7	16.2	152.6
4	.81	30.2	.62	22.7	8.3	111.8
3	.80	39.0	.37	37.5	3.9	67.9
	200 lb. Initial Pressure, by Gauge.					
70	.93	2.67	.76	17.7	135.8	143 9
60	.92	3.05	.80	16.7	125.7	152.1
50	.91	3.58	.84	15.9	112.4	59.6
40	.90	4.36	.89	15.1	97.6	168.9
30	.89	5.62	.94	14.2	80.2	178.9
20	.87	8.08	.99	13.6	58.4	187.4
14.7	*.85*	*10.60*	*1.00*	*13.4*	*45.2*	*190.2*
10	.84	14.86	.98	13.7	31.4	185.2
8	.83	18.15	.94	14.3	24.6	177.6
6	.82	23.42	.85	15.7	17.4	162.0
4	.80	33.74	.64	21.1	9.0	120.7
3	.79	43.56	.41	33.1	4.5	77.0

the diagram and increase F. In this way, too, reheaters and jackets (whose supply should be a charge against factor C, as not appearing as steam at high pressure cut-off) re-evaporated water and increase area of indicator diagrams. In some instances F becomes greater than unity; see the case of the single-valve auto-

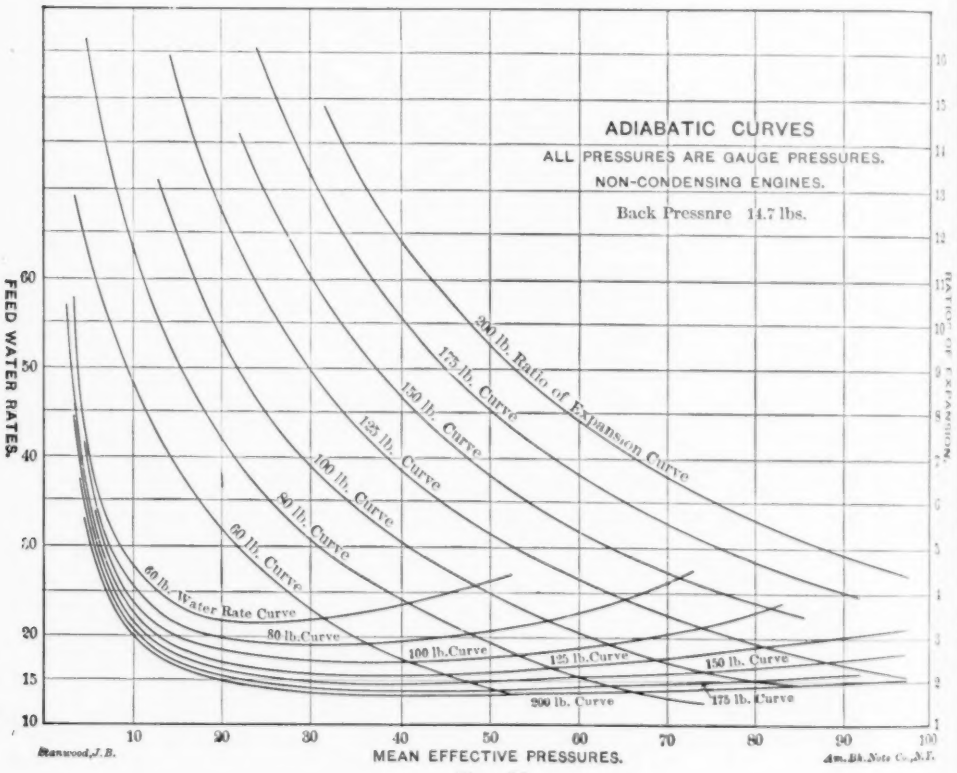

FIG. 53.

matic engine when operated under light load, tested by Mr. Armstrong, here F = 1.11. The writer has found in practice a number of such cases in which $F>1$, because, as already pointed out, with early cut-offs the actual terminal has been raised far above the adiabatic terminal in consequence of excessive re-evaporation or leakage, or both.

24. In Mr. Barrus' tests of a twin Corliss simple engine No. 31, F increases to a maximum value of 0.945 at short cut-off, then with still earlier cut-offs it decreases to 0.62. This is due to the

loop at toe of diagram. Mr. Barrus has kindly furnished copies of the diagrams taken from this engine, and Fig. 54 is a reproduction of the one having the largest $F=0.951$.

25. This diagram is one of the series where $E=0.673$; this is not the most economical test, as the cut-off is so early that the loss in C is enough greater to offset the high value of F. A later cut-off with F reduced to 0.92 produces a higher combined effect of CF, so that $E=0.685$. The terminal pressure in Fig. 54 is 18

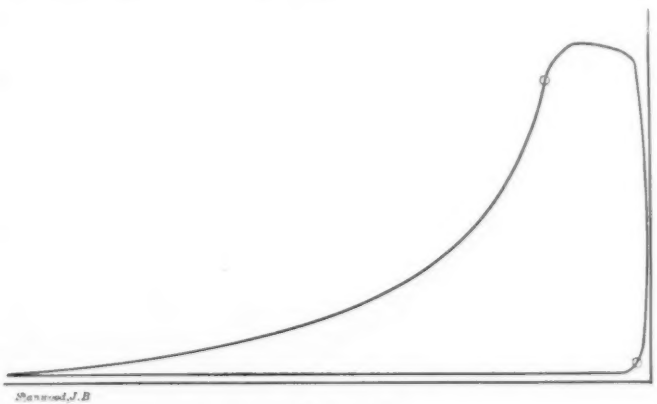

FIG. 54.

pounds absolute, back pressure 3 pounds; in the test yielding highest efficiency the terminal is about 24 to 25 pounds absolute. The terminal pressure for highest efficiency in single valve automatic series is between 36 and 40 pounds absolute; Mr. Willans claims with his engines the terminal associated with highest efficiency to be about 19 to 21 pounds.

The Condensation Factor "C."

26. The conditions that affect C are not capable of close analysis. Engineers know that condensation losses are large when temperature range is large; when clearance surfaces are extensive in proportion to volume of steam admitted; when revolutions per minute are small; when cut-off is early; and some believe (Mr. Willans especially) that entrained water retained in the cylinder each stroke is an important factor in increasing the value of $1-C$. There are, however, no data nor is there any theory to indicate

how much each of these conditions affects the total result. In this factor, as originally assumed, there must also be included the leakage losses which occur through the exhaust valves and piston before high pressure cut-off takes place, the leakage through the admission valves after cut-off occurs and the radiation loss from the cylinder during the period of admission.

27. On the other hand as tending to reduce 1—C there is the unknown effect of wire-drawing during admission which may superheat slightly the admitted steam, or at least dry out some of

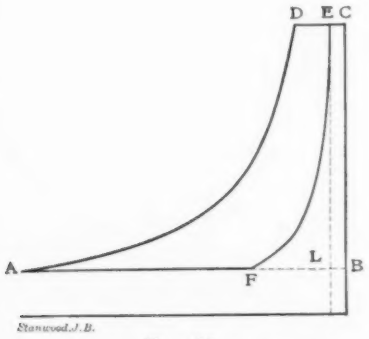

FIG. 55.

the entrained water associated with that steam; this water should not properly be charged to the engine, however, but should be deducted from the feed supplied.

28. These conflicting, varied and uncertain conditions, which only the combined skill of the designer, builder and perhaps the operator can keep under control, produce a resultant with which we are all familiar and which we include in the general expression "cylinder condensation."

The Effect of Clearance.

29. Clearance affects the condensation factor, C, indirectly and the expansion factor, F, directly. The result of clearance in reducing the actual ratio of expansion is well known; conversely if a given actual ratio of expansion is required, the greater the clearance the earlier the cut-off must be to secure it. This condition is well illustrated by two diagrams submitted to the Com-

mittee on Standardizing Engine Tests, by Mr. Frank Ball, one of our members.

30. They are reproduced in the accompanying Figs. 55 and 56, in which $F = 1$ in both cases; but with the smaller clearance the cut-off is about $\frac{1}{9}$, with the larger it is $\frac{1}{18}$; and the mean effective pressure is also nearly twice as large with the smaller as with the larger clearance, while the *ratio of expansion* remains the same. Therefore to secure given values of F, large clearances require larger cylinder capacity, earlier cut-offs and smaller mean effective pressures than small clearances; all of these conditions tend indirectly to induce heavier initial condensation. In Table III there is evidence of this taken from actual practice; in two series of tests

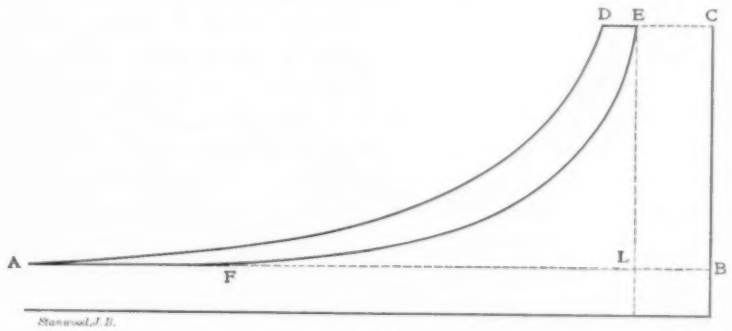

FIG. 56.

made with 100 pounds steam pressure, one on a simple twin Corliss engine No. 31 having small clearance tested by Mr. Barrus, the other on a simple single valve automatic engine having larger clearance tested by Mr. Armstrong, selected so that F is practically the same in both, the following relation of cut-off, cylinder condensation and mean effective pressure is revealed:

Corliss. . . . . $F = 0.92$, Cut-off 0.231, Cyl. condensation = 0.255, m.e.p. = 33.1 lb.
Single-valve. $F = 0.901$, Cut-off 0.130, Cyl. condensation = 0.380, m.e.p. = 19.0 lb.

31. As to the direct effect of clearance on F, the raising of terminal pressure by the employment of large clearances is generally understood; as a result there is always free expansion without work whenever release pressure exceeds back pressure; the same free expansion takes place when compression pressure does not rise to initial pressure. Both conditions evidently reduce the value of F directly, and are affected by the degree of clearance.

The Effect of Compression and Release.

32. Some of the effects of compression on factor F have already been seen in connection with clearance; indeed the proper treatment of clearance spaces is closely related to the correct use of compression. With no clearance, compression would be impossible; so with engines of high rotative speed, when compression is used as a means for cushioning the reciprocating parts, it becomes necessary to supply sufficient clearance into which "cushion steam" may be compressed.

33. With release, considered independently, the chief object is to secure a sufficiently early opening so that the steam may be evacuated from the cylinder without back pressure. In engines where load is uniform, and terminal pressure is always near to the back pressure, the release may be quite late. Whenever heavy overloads with high terminals are often maintained, then the release should be earlier, as free expansion from a high terminal gives an increased volume to be rapidly evacuated before the return stroke of the piston is undertaken. These conditions most desirable for release, cannot always be secured without an interference with compression, or vice-versa, as both events are controlled by one and the same edge of the exhaust valve. Therefore, with single eccentric Corliss engines, the range of compression is greatly restricted by the evils of too late a release; or with single valve automatic engines, as the cut-off becomes early the release increases abnormally with an increase also of compression.

34. In this last instance we have a state of affairs tending to reduce F, as a high premature terminal curtails the expansion and by a terminal drop increases the amount of free expansion. This creates a condition similar to that with clearance, due to what may be called a low "factor of evacuation," which favors an increase also of cylinder condensation, $1—C$. This factor, k, is the ratio of the evacuated volume of steam at release pressure per stroke to the volume displaced by the piston per stroke; and is illustrated in Fig. 52, by the ratio $\frac{VY}{SQ}$, and in Figs. 55 and 56, by the ratios $\frac{AF}{AL}$. As the actual ratio of expansion * in Fig. 52 is

* In estimating the "actual ratio of expansion" from actual indicator cards, the ratio of the volume of evacuated steam at *release* pressure to the volume of steam at *cut-off* pressure, both as accounted for by the indicator, has been selected, as such a ratio facilitates the discussion of losses.

$\frac{VY}{LX}$ it follows if k is made smaller and this certain ratio of expansion is to be maintained, an earlier cut-off must be used, and a smaller mean effective pressure will result, both contributing to a larger value for condensation loss $1—C$.

35. Mr. Willans avoided this difficulty in his high speed engine, by relieving the working steam cylinders of the duty of cushioning the reciprocating parts, assigning this duty to an auxiliary air cylinder or buffer. In this way he was enabled to reduce clearance, compression and release, so that k in his engine as tested was about 0.95 and large mean effective pressures for the grade of expansions used were secured by late cut-offs, so contributing to a low initial condensation.

36. In Corliss engines, at moderate speed, where but little cushion is required, k rarely falls below 0.90. If however higher revolutions per minute come more generally into vogue, double eccentrics are necessary to obtain sufficient release and compression and k is bound to diminish. With single-valve automatics k is often about 0.60 and with light loads far less.

37. By avoiding the terminal loop, large clearances with ample compression and early release may be more economical on very light loads, than small clearance with late compression and release. Comparing again the Corliss and single valve automatic tests, Table III, the latter are more economical when lightly loaded. Fig. 57 gives the curves of cylinder efficiencies E of these two engines for corresponding mean effective pressures; for mean effective pressures less than about 18 pounds, the advantage in this comparison evidently lies with the single-valve engine.*

The Effect of Varying Loads.

38. As a variation in the load thrown upon an engine calls for a corresponding direct variation in the mean effective pressure, it follows that the study of the possible cylinder efficiencies or water rates of different mean effective pressures small and large will be a profitable one. Especially is this the case since the modern method of directly driven electrical machinery has decreased the load factor under which engines operate. Compar-

* Mr. Henry E. Longwell points this out in a communication to the discussion on "Effect of Clearance of Steam Engines," Mr. Kingsbury's paper in *Transactions* of the Society, vol. xxiii, page 192.

ing an engine continually harnessed to the transmitting mechanism of a plant whether useful work is done or not, with one in which this mechanism is in operation only when useful work is done, it

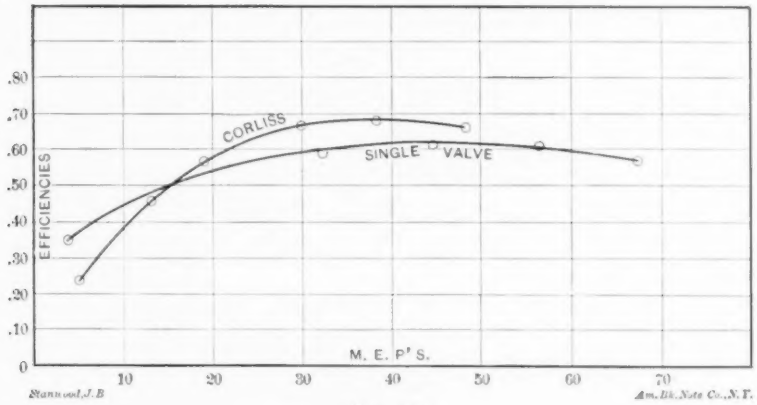

FIG. 57.

will be found that the range of mean effective pressure will be narrower with the former than with the latter. For example, Table V shows what the variation of mean effective pressure will

TABLE V.

EFFECTIVE LOADS.	DEAD LOAD.			
	15 per cent. of Full Load.		50 per cent. of Full Load.	
	M. e. p.	Total i. h. p.	M. e. p.	Total i. h. p.
One-quarter load	14.5	42.6	25	125
One-half load	23	67.6	30	150
Three-quarters load	31.5	92.6	35	175
Full load	40	117.6	40	200
One-quarter overload	48.5	142.6	45	225

be in two cases, where the dead loads are assumed to be 15 and 50 per cent. of full indicated power, with a variation in useful work from $\frac{1}{4}$ full load to $\frac{1}{4}$ overload; 40 pounds is assumed to be the mean effective pressure of each engine at full load, and the useful work done to be 100 horse-power.

39. When the dead load is small a much smaller engine can be used, but with such an engine when operated at light loads the

mean effective pressures will be smaller and less economical per horse-power than in case of engines with large dead loads.

40. Therefore if lighter loads prevail good practice has shown that to secure reasonably good results, and to bring the mean effective pressures for light loads to more economical conditions, it is advisable to secure as high a mean effective pressure for maximum loads as is possible. This desirability of having large mean effective pressures makes it important to know what the theoretical mean effective pressures are for different grades of expansion and initial and terminal pressures.

41. Table IV, giving values for adiabatic expansion, was calculated for this purpose. It gives for the different initial pressures 60, 80, 100, 125, 150, 175 and 200 by gauge, and for different absolute terminal pressures, p^1, the mean effective pressures theoretically attainable. In it are also given values of the quality of steam after adiabatic expansion, x; the ratio of expansion, R; the factor of complete expansion, F, which indicates the loss due to a terminal loop or a terminal drop; the theoretical water rate, S; and the theoretically available British thermal units, U_1. The mathematical basis of this table may be found in appendices B, C and D.

42. A chart, too, has been constructed from this table (Fig. 53) showing the relation between mean effective pressure, ratio of expansion, and water-rate for different initial pressures. The base line of the chart measures the mean effective pressure; the left hand vertical scale the water rates, and the right hand vertical scale the ratios of expansion. The curves representing water rates for varying mean effective pressures at different pressures, by gauge, are those of a hyperbolic type, at the left and lower part of chart; the curves representing ratios of expansion are the long sweeping ones extending from the upper left to the lower right hand corner. To find, for example, the theoretical mean effective pressure and water-rate for 4 expansions at 125 pounds pressure—on the line passing through 4 on the right hand scale—find its intersection with expansion curve for 125 pounds. Vertically below this, 63 mean effective pressure is found on the base line. Following up the vertical line passing through 63 mean effective pressure to its intersection with the water-rate curve for 125 pounds, 17 + pounds will be the water-rate desired.

43. The ratio of the mean effective pressures of practice to the theoretical mean effective pressures of the table at different ex-

pansions may be expressed by a diagram factor, d, which is made up of perhaps three other factors; one of these is the factor of evacuation, k, whose influence has already been noted. For instance by table IV the theoretical mean effective pressure for an engine without clearance, with 100 pounds initial pressure, expanding 4.62 times to a terminal of 20 pounds *abs.* is 43.5 pounds, but if by assuming a clearance so that (all other diagram losses neglected) $k = 0.9$, the mean effective pressure will be reduced to $0.9 \times 43.5 = 39.2$ pounds; or, if $k = 0.6$ then mean effective pres-

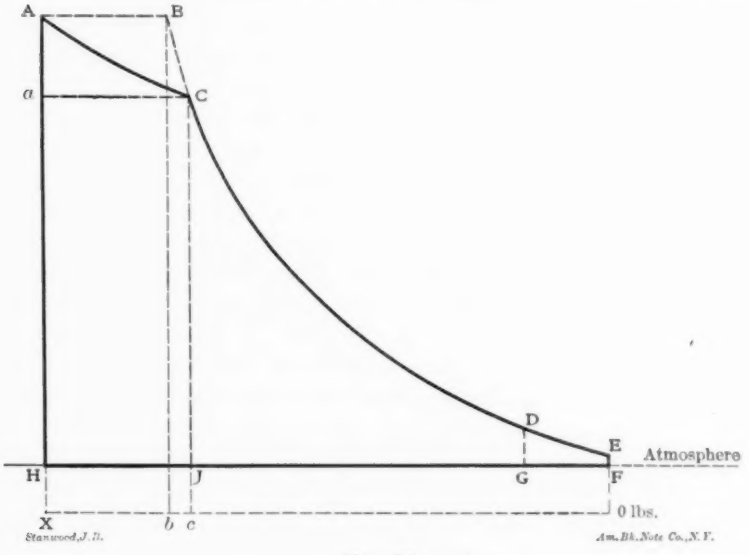

FIG. 58.

sure = 26.1 pounds. This diminution may occur without in any way affecting the value of F, because the release, compression and cut-off may be reduced in such proportions that the initial and terminal pressures remain always constant.

Another factor of d is one that may be called the wire-drawing factor, g, it gives the reduction of the theoretical diagram as a result of throttling, or pressure reduction during the period of admission; usually it does not affect either the cut-off, compression or release, and has but a slight influence at ordinary loads on F. Fig. 58 shows the effect of wire drawing and the degree of reduction in the mean effective pressure as expressed by g; here $ABDGH$ represents the mean effective pressure for, let us say,

y expansions from an initial AB to a terminal pressure at D, AB representing the volume of one pound of steam at initial pressure. If the cut-off pressure is reduced by wire-drawing to C, so that aC now equals the volume of one pound of steam (nearly) at cut-off pressure, then with y expansions the terminal pressure drops to EF, which is lower than DG. The area $ABDGH$ approximately equals $ACEFH$, for ABC will nearly equal $DEFG$ so that the factor F* is but slightly affected. By inspection it will however be found that this modification of the diagram reduces the mean effective pressure because the mean effective pressure $ACEFH$ measured on HF is less than that of $ABDGH$ on HG, and smaller by an amount of some ratio varying about as $\frac{Ce}{Bb} = g$.

44. The third factor m, made up of minor deficiencies in the diagram, such as free expansion in clearance spaces, back pressure, wire-drawing at release or compression, and in compound engines losses due to free expansion and wire-drawing in receiver and connecting pipes, cannot be definitely determined, but its direct effect is to reduce the value of F. In simple engines m is smaller than in compound engines as there is no chance for intermediate losses between the cylinders. Algebraically we can express the resultant of these factors as $d = kgm$. The following Table VI gives the values of these quantities for three simple and two compound engines, as tested, assuming $g = \frac{Ce}{Bb}$.

TABLE VI.

DIAGRAM FACTORS AND COMPONENT PARTS FROM PRACTICE. $d = kgm = \frac{M_1}{M}$.

ENGINE TESTED.	Actual M. e. p. M_1.	Theoretical M. e. p. M.	Diagram factor $d = \frac{M_1}{M}$.	k.	g.	m.
Corliss No. 1.....	33.12	51	.65	.91	.77	.93
Corliss No. 2.....	41.5	55	.75	.97	.78	.99
4-Valve Gridion type, 20b.....	24.0	31	.77	.93	.87	.96
Single Valve, Compound, 54....	33.8	72	.47	.61	.86	.90
4-Valve, Compound, 46B.....	34.4	52.2	.66	.93	.77	.93

45. The indirect effect of a small value of d on the condensation loss, 1—C, is to increase it, for the tendency with a small diagram

* F may be seriously affected by initial wiredrawing if the terminal is so low as to form a loop, while if the initial pressure is maintained to point of cut-off the loop is prevented.

factor (for any desired degree of expansion) is to cause the cut-offs to be earlier and the cylinders larger than otherwise would be the case, with consequent increase in cylinder condensation per volume of steam used.

46. From this general survey of conditions affecting C and F it appears that the one great contributing cause to an increase in cylinder condensation, $1—C$, is what may be called a conspiracy on the part of most systems of steam distribution to reduce the amount of mean effective pressure that should theoretically be secured per given ratio of expansion; as a result, earlier cut-offs and larger cylinders have to be used to secure the degrees of expansion necessary for economy than otherwise would be the case if more nearly theoretical conditions obtained.

---

47. In conclusion: as modern practice exacts the employment of smaller mean effective pressures than formerly on account of greater variations in load, it becomes interesting to examine the performance of different types of engines and note which give the highest economy with light loads or small mean effective pressures.

48. If $\frac{1}{4}$ delivered full useful power with a dead load 15 per cent. of full load be taken as the lightest worthy of consideration, then 14 pounds mean effective pressure represents the corresponding mean effective pressure for a full load of 40 pounds. Table VII gives such results as are usual with simple Corliss and single-valve engines and compound single-valve engines with about this mean effective pressure.

TABLE VII.

USUAL WATER RATES, ETC., AT 14 POUNDS M. E. P.

TYPE.	Water rate about.	Boiler Pressure.	R. p. m.
	Lb.	*Lb.*	*Lb.*
Corliss, Simple	35	100	85
Single Valve, Simple	35	100	280
Single Valve, Compound	29	130	270

49. This shows at a glance the advantage of the single-valve compound engine for this load; but these values should be considered in connection with the average conditions of maximum economy and capacity, see Table VIII.

TABLE VIII.

USUAL MAXIMUM ECONOMIES AND CAPACITIES.

TYPE.	MAXIMUM ECONOMY.		MAXIMUM CAPACITY.	
	Water-rate about.	M. e. p.	M. e. p.	Boiler Pressure.
	Lb.	*Lb.*	*Lb.*	*Lb.*
Corliss, Single (single eccentric)....	26	40	57	100
Single Valve, Simple.....	30	45	67	100
Single Valve, Compound.....	23	36	42	130

50. From these exhibits it appears that the single-valve compound leads except as regards one point, viz., that of capacity; and this to so great an extent that if designed for $\frac{1}{4}$ overload, the mean effective pressure for $\frac{1}{4}$ load falls below 14 pounds, and the economy is not much better than the other at 14 pounds. If this type of engine can be developed so that its capacity can be increased without a corresponding interference with its other good qualities, its general simplicity warrants its becoming a commercial success. That there is a chance to secure reasonably good economy with small ratios of expansion for overloads is shown by Table IV and Fig. 56. There is also ample room for increasing the efficiency, especially if a reduction in the largest loss—cylinder condensation—can be effected; large diagram factors, d, are instrumental in helping this reduction.

APPENDIX *A*.

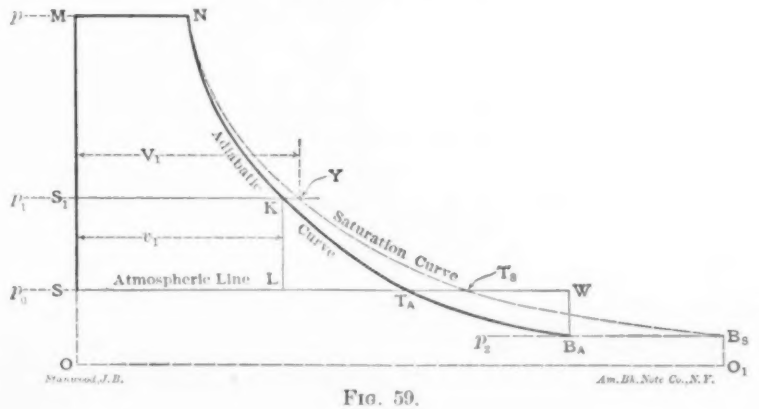

FIG. 59.

OO_1 = absolute pressure; SW = atmospheric pressure, or p_0.

For any pressure, p_1, $x = \dfrac{v_1}{V_1} = \dfrac{KS_1}{YS_1}$ = quality of steam.

For p_0, atmospheric pressure $x = \dfrac{T_A S}{T_s S}$

MNT_AS = Area of theoretical diagram, $F = 1$.
KT_AL = Loss in area when terminal $p_1 > p_0$.
T_AWB_A = Loss in area when terminal $p_2 < p_0$.

APPENDIX *B*.

The quality of steam after adiabatic expansion is shown on Entropy diagram (Fig. 60) by $x = \dfrac{a_1 r + ab}{a_1 b_1}$ (see A. S. M. E. *Proceedings*, Volume XXVI., page 757). But $N_s = yb$, $N_{s1} = y_1 b_1$,

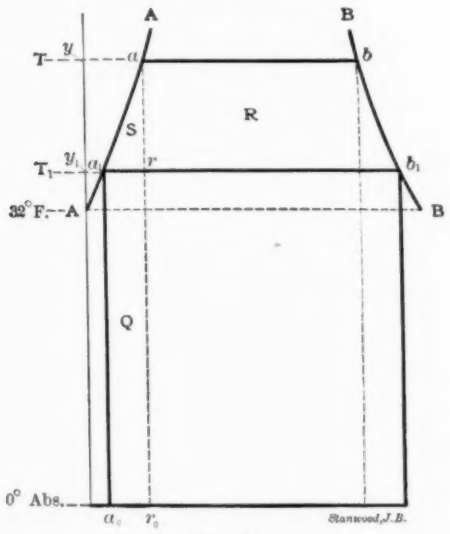

FIG. 60.

$N_w = ya$ and $N_{w1} = y_1 a_1$; therefore, $a_1 r + ab = N_s - N_{w1}$, and $a_1 b_1 = N_{s1} - N_{w1}$

$$\therefore x = \frac{N_s - N_{w1}}{N_{s1} - N_{w1}} = \frac{v_1}{V_1} \text{ of Fig. 60, Appendix } A.$$

APPENDIX *C*.

U = Area $R + S = R + (S + Q) - Q$, (Fig. 60), but $D = T - T_1$ and in British thermal units, D = area $a_1 a r_0 a_0$; $R = D\ (N_s - N_w)$; $S + Q = D$; $Q = T_1\ (N_w - N_{w1})$ $\therefore U = D\ (N_s - N_w) + D - T_1\ (N_w - N_{w1}) = D\ (1 + N_s - N_w) - T_1\ (N_w - N_{w1})$.

APPENDIX *D*.

If expansion does not reach back pressure, but p_1 a terminal greater than p_0, the back pressure (Fig. 59, Appendix *A*), then to value of U of Appendix *C* must be added the work, u, in British thermal units done by area $(p_1 - p_0)\ v_1$, or $u = \frac{144}{778}\ (p_1 - p_0)\ v_1$; and $U_1 = U + u$ (U being calculated for temperatures of p and p_1). But $v_1 = V_1 x = \frac{x}{w_1}$, w_1 = weight of 1 cubic foot steam at p_1 pressure.

$\therefore u = \frac{(p_1 - p_0)x}{5.4 w_1} = \frac{x}{W_1}\ (p_1 - p_0)$. $W_1 = 5.4\ w_1$. See Table I.

Therefore, $U_1 = D\ (1 + N_s - N_w) - T_1 (N_w - N_{w1}) + \frac{x}{W_1}(p_1 - p_0)$.*

U_1 in Table IV. has been calculated by this formula.

APPENDIX *E*.

To determine from any indicator card U_m the actual British thermal units converted into work per pound of steam accounted for by indicator at high pressure cut-off, divide the total work done per stroke by the weight of steam per stroke accounted for at high pressure cut-off. Let M = mean effective pressure, or if compounded the referred mean effective pressure; L = stroke in feet; A = area in square inches of high pressure piston; w = weight of 1 cubic foot of steam at high pressure cut-off pressure; $W = 5.4\ w$; b = fractional part of high pressure piston displace-

* Inadvertently the factor x was omitted in the communication by the writer to Committee of Standardization of Engine Tests—see formulæ 1 and 3, pages 815 and 816, vol. xxiv, American Society of Mechanical Engineers Proceedings.

ment occupied by the steam accounted for by indicator at high pressure cut-off pressure; r = ratio of high to low pressure piston areas.

Then total work in British thermal units per stroke $= \frac{MLA}{778} \ldots (a)$

and weight of steam per stroke at high pressure cut-off $= \frac{bLA\,w}{144} \ldots (b)$

(a) divided by (b) or $U_m = \frac{M}{5.4\,wb} = \frac{M}{Wb}$, or if compounded $U_m = \frac{M.r}{Wb}$

APPENDIX *F*.

To determine F from any indicator card, $F = \frac{U_m}{U}$ or $\frac{M.r}{UWb}$, U being determined by Fig. 51 or Table II. F, also $= \frac{S_m}{S}$ or water rate as calculated at high pressure cut-off divided by minimum theoretical water rate of Fig. 51 or Table II.

APPENDIX *G*.

To determine mean effective pressure for adiabatic expansion, it is first necessary to determine the foot-pounds of work done per stroke in an engine using 1 pound of steam per stroke. If this steam be supposed to be acting in a cylinder whose piston area = 144 square inches, then the volume of adiabatically expanded steam in cubic feet equals the stroke in feet. Now the foot pounds of work per stroke $= 778\ U$; the volume of expanded steam $= x\,V_1 = \frac{x}{w_1}$; if M = mean effective pressure, then $144\,\frac{Mx}{w_1} = 778\ U$ and $M = \frac{5.4\,w_1 U}{x} = \frac{W_1 U}{x}$.

No. 1063.*

CENTRIFUGAL FANS.

BY AUGUST J. BOWIE, JR., SAN FRANCISCO, CAL.

(Non-Member.)

1. The design of centrifugal fans has a field peculiarly its own, and while it is true that the same general laws govern the design of centrifugal pumps and fans, still the difference in the nature of the fluids, air and water is so great that a design suitable for one cannot often be used to advantage for the other.

2. Losses of energy are of course inseparable from any form of mechanical apparatus, but the proper and most efficient design consists in proportioning properly these losses and not allowing any one to become excessive.

3. The losses in a centrifugal fan may be classified in general as follows:

(1.) Loss due to a sudden change of velocity of the air.
(2.) Loss due to sudden change of direction of the air.
(3.) Loss due to air friction.
(4.) Loss due to back slippage of air past the running rings.
(5.) Loss due to eddy currents.
(6.) Mechanical friction in the bearings.

4. *Loss* 1. Sudden change of velocity. When the velocity of a body of air is suddenly increased or decreased there will be a loss of head, and consequently of energy, due to impact. The loss of head due to this cause will vary as the square of the speed of the air, provided the ratio of velocities be constant. Sudden change of velocity of any magnitude should be avoided as much as possible.

5. *Loss* 2. Sudden change of direction. This loss is similar to the previous one, and should be particularly avoided where the

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

velocity is great or where uniformity of velocity is desired for other reasons.

6. *Loss* 3. Air friction. Loss of head due to this cause will vary as the square of the speed. In a properly designed fan the losses due to the first three causes will be so proportioned that their sum is a minimum. This determines in part the design of the casing. Of course the casing should be as smooth as possible, and all parts of the fan should be so constructed as to offer a minimum in the nature of obstructions to the flow of the air.

7. *Loss* 4. Back slippage of air. This is a local circulation of air within the fan and casing. This loss may be cut down by reducing the area past which the air can circulate. This means a reduction of the mechanical clearance of the running parts of the fan. In the case of a closed runner the running rings should be so constructed as to render it difficult for the air to circulate past them. With centrifugal fans considerably more clearance is required than with centrifugal pumps, where water lubrication of the running rings allows the use of very small clearances.

8. *Loss* 5. Eddy currents. In some fans this is quite a serious source of loss. To avoid this loss a thorough understanding of the action of air in motion is decidedly necessary.

9. *Loss* 6. Bearing friction. This loss is of two kinds: First, that due to the natural friction of the dead weight of the fan, and secondly, that due to lack of balance of the fan. The loss due to the latter cause is apt to be very serious, owing to the high velocities of revolution, and to impair very materially otherwise excellent results. It is perhaps needless to add that the losses due to bearing friction depend very much on lubrication of the bearings, and that self-oiling bearings should by all means be used on such high-speed machinery. This is particularly true if there is any unbalance of the runner, when the bearing friction may be a comparatively large proportion of the total input.

A study of the flow of air is of great assistance in fan design. To attempt to make air flow in certain fixed paths without due consideration of whether it can be forced through those paths at a uniform velocity is certainly not the right way to approach the subject. The usual result of such a procedure is that the flowing air will not fill the channels through which it must pass, thus resulting in a very unequal distribution of velocity, and consequent loss in eddy currents and in other ways.

10. A graphical plot of the lines of flow and the velocities of

the same is perhaps the best method of studying the action of air in motion. To obtain results suitable for making such a plot an anemometer is practically out of the question, being entirely too large to use for any such purpose, besides being open to other objections.

11. By means of a Pitot tube rough approximations may be made of the velocity and direction of flow at any point. However, the pressures obtained will, in many instances, be so complicated by an existing state of pressure of the air as to render the readings of the gauge of little value as a means of indicating the actual velocity of the air. This, too, in cases where it would seem

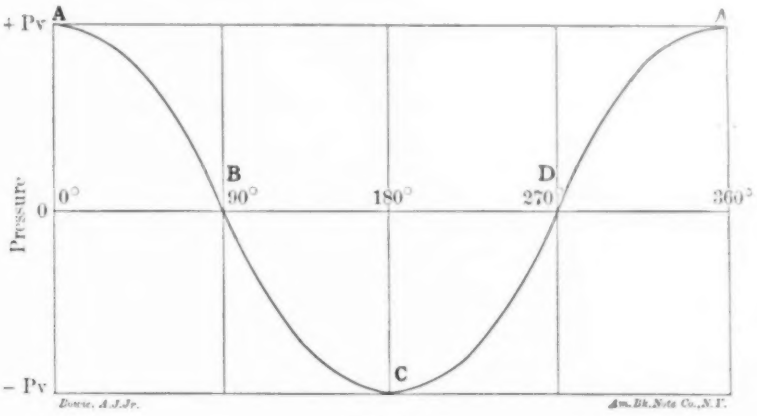


FIG. 61.

almost incredible that a state of pressure different from that of the atmosphere could exist.

12. With the idea of obtaining a suitable velocity gauge for air, which would be independent of any existing state of pressure, the writer designed the following apparatus. Two small brass tubes were soldered together side by side. Openings were made in these tubes, directly opposite to each other, 180 degrees apart. In order to have the holes as nearly opposite in direction as possible, they were formed by soldering a small piece of tube on the end of each tube, so as to form a kind of right angle turn. These pieces were then filed down so as to make the ends of the tubes practically no larger than the remainder of the tubes. Rubber tubing connected the other ends of the brass tubes, to the two

sides of a sensitive differential pressure gauge. The brass tubes were mounted so as to turn about their axis, in a suitable head provided with a scale, laid off in degrees. The tubes themselves carried a pointer for the scale.

13. The idea of this apparatus was as follows. If a small tube, with an opening in one side, be placed in a body of fluid, flowing at a uniform velocity, normal to the tube axis, then the resulting pressure in the tube will be the sum of the static pressure of the fluid, and of the dynamic pressure of the fluid, impinging on the opening. According to theory, the dynamic pressure will vary from the pressure due to the velocity head, Pv, to $-Pv$, as the

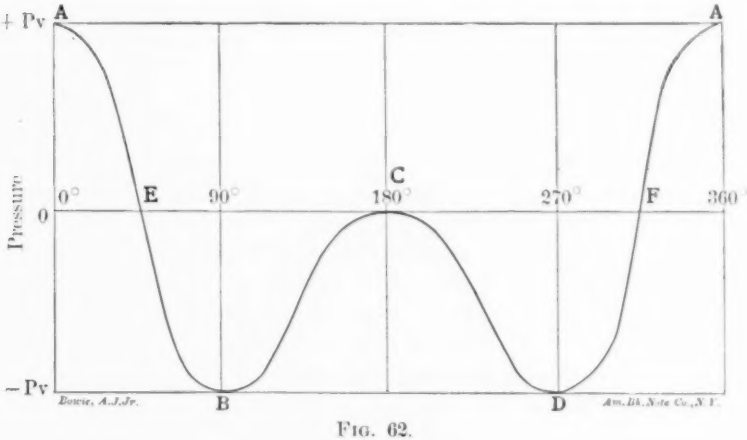
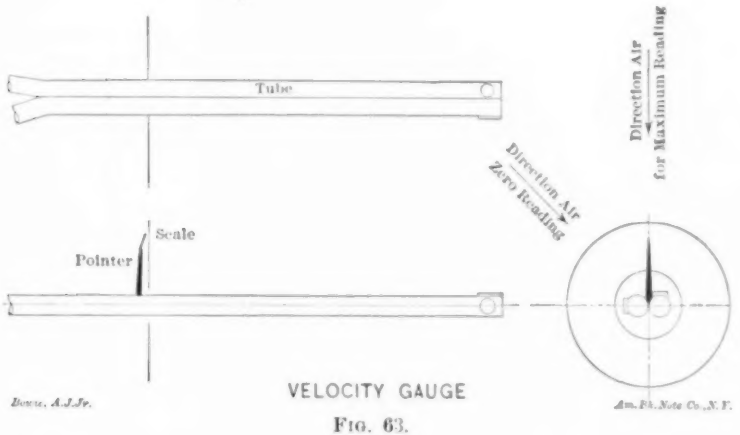


FIG. 62.

tube is revolved; being $+Pv$, when the opening is directly opposed to the flow of fluid; O, when the opening is 90 degrees from this position, and $-Pv$, when the opening is 180 degrees from the first position.

14. The tubes were to be placed in the path of flow of air, and normal to the same. The static pressure would be equalized in each of the tubes and the pressure difference would be due to dynamic pressure alone. This would vary as the tubes were revolved; being a maximum when the opening in one was directly opposed to the flow of air, and the other tube was in consequence subjected to the negative pressure due to the velocity of flow. Calling the static pressure Ps, the pressure in the first tube would be $Ps + Pv$, and the pressure in the second tube would be $Ps -$

Pv. The pressure difference would be 2 Pv, — a maximum. From this the velocity could easily be calculated. As this maximum point of pressure differences would be at the flat portions A and C of the pressure curve, theoretically obtained by revolving a Pitot tube in a stream of air (see Fig. 61), it would be difficult to ascertain, at all accurately, the direction of flow therefrom. Hence it was intended to use a zero method to attain this end. In other words, the direction of flow of air, was to be found by equalizing the pressures in the two tubes. This point of equal pressures would occur at BD, when the line of the openings in the tubes was normal to the line of flow of the air.



VELOCITY GAUGE

FIG. 63.

15. However, when the zero method was tried, the results were rather indefinite and unsatisfactory. In order to see the reason for this a test was made, utilizing only one side of the double tube, and opening the other side of the differential pressure gauge to the atmosphere. The tubes were then revolved, and pressure readings and corresponding angular readings were taken. The result of this test is shown in the curve Fig. 62. The curve is in the nature of a W, having two maxima and minima. The point of maximum pressure is when the opening is directly facing the stream of flow. The points of minimum pressure are at right angles to this line of flow, while at 180 degrees from this line of flow is another maximum point. It has usually been considered that the last mentioned point was a minimum, and that the points where the open-

ings were normal to the line of flow, would give readings independent of the velocity of flow. Instead of this, we find that the conditions are approximately that the point where the opening in

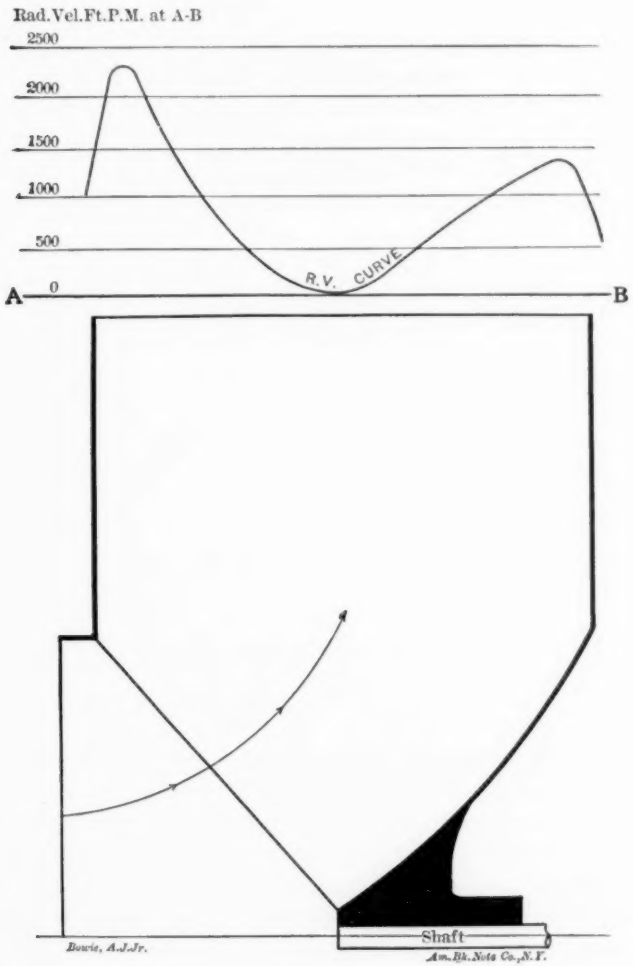


FIG. 64.

the tube is opposed to the flow, will give a pressure $Ps + Pv$; the 90 degree points give $Ps—Pv$; while the 180 degree point gives Ps. This statement is a rough approximation, and will, of course, vary somewhat with the form of the tube. The reason

why the apparatus had failed to work accurately on the 90 degree points, is apparent. *B* and *D* (Fig. 62) were these points, and except for the fact of being a zero method, it would have been

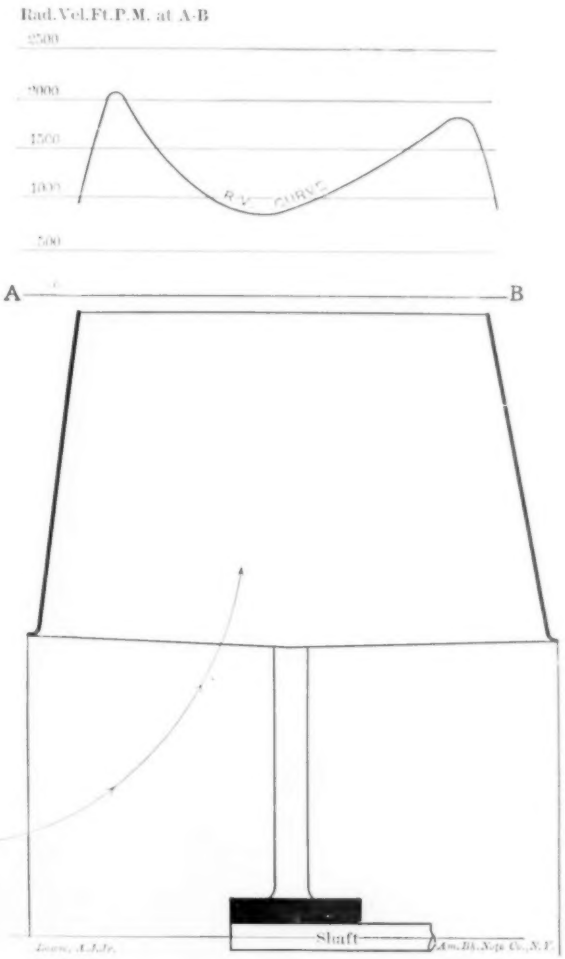


FIG. 65.

impossible to have found two more unfavorable points for a differential reading.

16. Next, the two tubes previously used, were shifted around 90 degrees, so that the two openings were at right angles to each

other, instead of being 180 degrees apart. (See Fig. 63.) The effect of this was that a maximum reading was obtained at AB, or AD, and a zero reading at EF. This maximum reading was about twice

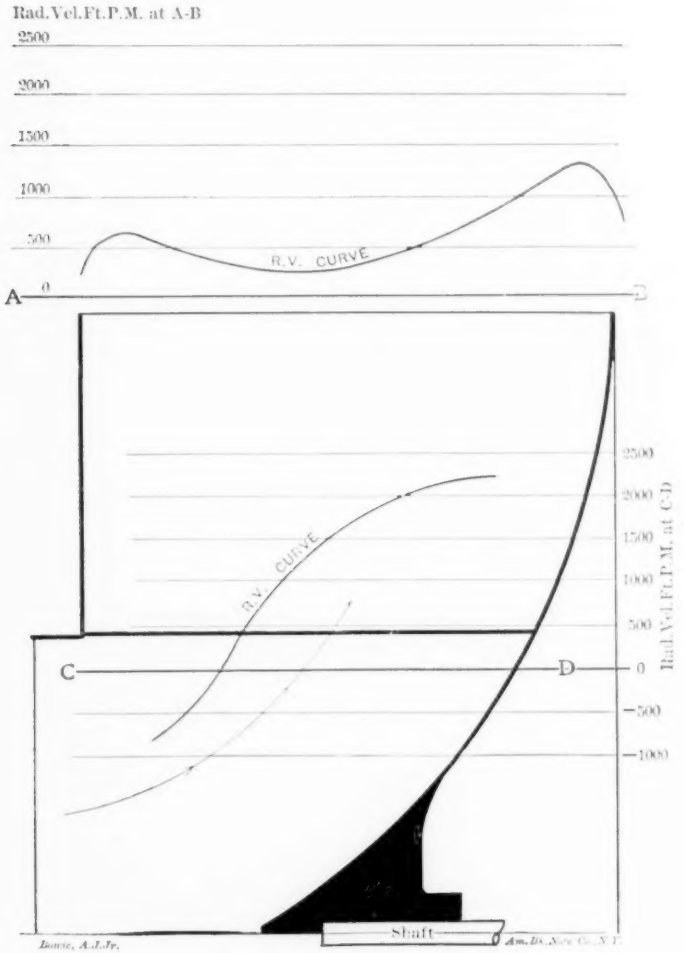


FIG. 66.

as large as had been previously obtained; and the zero reading was at the most sensitive point on the scale, the slightest variation of angular position of the tubes making one pressure go up and the other fall rapidly. The gauge, as altered, became so exceedingly

sensitive, that with the comparatively crude apparatus used, it was an easy matter to get the correct angle of flow of the air within ½ degree. The gauge was calibrated both for angular position, and for pressure, by placing the tubes in a stream of air, flowing with a known velocity, and in a known direction.

17. With the aid of this apparatus, it was a comparatively simple proposition to determine exactly what was going on at any point in an air path not cut by moving vanes. By boring a small hole in a fan casing close to the periphery of the fan, it became possible to tell just how the air was leaving the runner of the fan, both as regarded velocity and direction. By mounting the tubes and head in suitable guides, readings could be taken across the face of the fan, and the surface distribution of air plotted out. In order to get the actual delivery at any point in the face of the fan, the velocity of air at that point should be multiplied by the sine of the angle that the direction of flow makes with the tangent to the circumference of the fan at the point in question. By the proper use of this apparatus, the presence of eddies may be discovered, and suitable provision may be made in the design of the fans, to eliminate them.

18. The writer has obtained some very peculiar results by applying this method of testing to obtaining the radial discharge from fans; and the radial discharge is the true criterion of the amount of air actually delivered from the periphery of the fan. In one case, a certain single suction fan, when run beyond its capacity, gave all the discharge from the sides of the runner; the air in the center of the face of the runner was actually on the point of starting to go the other way. This was an example of bad distribution of air.

19. A uniformly distributed flow of air is, of course, conducive to better results, though some fans show a fair efficiency in spite of poor air distribution. Waste space is a thing to be avoided in fan design. It serves merely as an aid to eddy currents, with their inevitable losses.

20. Figures 64, 65 and 66 show some results obtained with various runners. The figures show half sections of the runners, and the radial velocities of discharge from the periphery of the same. Owing to the crudeness of the apparatus used, and to the fact that the tubes were of necessity some short distance from the periphery of the fan, to allow running clearance, the results are not strictly accurate. Still they show quite clearly some rather re-

markable distributions of air. Fig. 66 shows in addition to the peripheral radial discharge the radial velocity of the air entering the fan, manifesting a very unequal distribution of velocity. In fact, the air is entering radially through only about two-thirds of the runner, and is actually flowing back through the other third.

DISCUSSION.

Prof. R. C. Carpenter.—As I have been making experimental investigations of centrifugal fans to greater or less extent during the past ten years, I read the paper which Mr. Bowie has written with a good deal of interest; but I am very sorry to say that I cannot find that he has added in any way whatsoever to our stock of knowledge regarding this important class of machinery.

I am not certain that I disagree with any of the very general propositions which are stated, as they are of the fundamental kind which are evident to any student of the subject. In paragraph 19 the statement is made, "waste space is a thing to be avoided in fan design." If Mr. Bowie means by this statement that a large clearance space is not desirable between the fan-wheel and the casing, I should most emphatically disagree with him, for the results of an extended series of experiments have shown that clearance space between the casing and the fan-wheel is of great importance. In a large measure the efficiency of the centrifugal fan increases with the form and capacity of such clearance space. The form of the casing and the form of discharge pipe are matters of great importance, practically as well as theoretically.

In paragraph 11, the statement is made that "by means of a Pitot tube a rough approximation may be made of the velocity of flow at any point." He then proceeds in the next paragraph to describe as a new and "suitable velocity gauge for air" an exceedingly crudely constructed Pitot tube, with which he obtains some results which are novel, to say the least, and are hardly explained by anything in connection with the paper.

I object especially to the statements that the Pitot tube is an instrument suited only for rough approximation in this work, and also to the implication that the tubes shown as something new are not forms of the Pitot tube, and that they are essentially different from those previously used by other investigators.

The accuracy of the Pitot tube for measuring water has been well established by exhaustive experiments on many forms of nozzles, and also in connection with various differential gauges. See paper by Williams, Hubbell and Fenkell, Trans-American Society Civil Engineers, 1902; also paper by W. M. White, 1901, Association of Engineering Societies, and by W. B. Gregory, A.S.M.E., Vol. XXV. Trans. Am. Soc. Mech. Engineers. The accuracy of this tube for measuring air was established by the Prussian Mining Commission in 1884, who calibrated various measuring appliances by aid of a large gasometer which contained 70,000 cubic feet, and I have used it extensively for purposes of measuring the velocity of air.

Mr. William E. Crane.—Are any of the members familiar with the Sirocco fan? As I understand the paper it alludes to the fan as ordinarily planned. The Sirocco fan is a new affair and does not have the large fan blades. It has blades something like the blades of a steam turbine, about an inch wide, and the hollows of these blades run towards the wind, that is, the wind is driven from the hollow. They require a great deal less power, and, with the same power will deliver a great deal more air.

*Mr. Aug. J. Bowie, Jr.**—Professor Carpenter has stated, with reference to the curves shown in Figs. 64, 65 and 66, that they are not explained by anything else in my paper. I fail to understand how any one who has read the paper carefully can find these curves to be anything except a perfectly logical conclusion of the paper, the main part of which deals with the study of the flow of air, including the particular form of apparatus found to be most advantageous for investigating the same. This is followed by the practical application of the method, to ascertaining the conditions of flow at entrance to the runner vanes and at exit therefrom. The conclusions drawn from the tests showed certain points which should be avoided in fan design; some of the objectionable points in the three designs of runners shown being plainly brought out.

I began my paper by studying the six laws on which depend the efficient design of fans. As Professor Carpenter has stated, they are of the fundamental kind. Still, as Professor Carpenter has made such a thorough study of fans, he must be aware that most of the fans built here are constructed in direct violation of most

* Author's Closure, under the Rules.

of these fundamental laws. Possibly this is due in part to a desire to cheapen the fans, and from this point of view is certainly successful. Still the result is obtained at such a sacrifice of efficiency that were a purchaser aware of the same, he would not hesitate to pay a much higher price for a higher efficiency fan. Since this state of affairs exists, these fundamental laws are certainly worthy of mention.

With regard to the statement that waste space was a thing to be avoided in fan design, I referred, as would naturally be inferred from the context, to waste space in the path through which the air must pass in its journey through the fan casing. This is well exemplified in the fan in Fig. 66, where the entering air is flowing back into the fan through about one-third of the space, where the air should be entering the runner. This space is a positive detriment, and serves only as a source for eddy currents. I was under the impression that the paper had brought out clearly the fact stated in the last part of Paragraph 9, which, summed up, means the following. To attempt to make air flow in certain paths without due consideration of whether it can be forced through those paths, at uniform velocity, will usually result in the air not filling the channels of flow, and, hence, in an unequal distribution of velocity, with consequent eddy current lost. The three fan designs shown clearly represent poor velocity distribution.

In Paragraph 11, I made the statement that by means of Pitot tubes, rough approximations may be made of the velocity and direction of flow at any point. This statement was intended to be general, and not to apply in the narrow sense in which Professor Carpenter takes it. I by no means meant to state that under certain conditions the Pitot tube cannot be used as an accurate velocity gauge, but that in the general case, where a state of pressure existed to complicate matters, its reading would be of doubtful value, unless suitably calibrated. For example, I took a regular Pitot tube and revolved it in a current of air flowing uniformly through a pipe, and obtained results similar to those shown in Fig. 62, and not like the results of Fig. 61. Consider, for example, the case of determining the velocity and direction of air flowing from a runner within the casing. The air discharged from the periphery of a runner is discharged with a certain velocity, and is also under a static pressure in the casing. To attempt to determine the velocity and direction of the air with a

single Pitot tube, would be a laborious and uncertain proceeding, as is apparent to one who has studied the subject. A double tube and differential gauge were obviously a great improvement over a single tube. I fail to understand from what Professor Carpenter draws his inference as to my implication that the device I have used is not a form of Pitot tube. The fact of fastening together two Pitot tubes seems hardly necessary to explain is an adaptation of the Pitot tube. The use of a differential gauge in this connection is so old that it seems hardly fitting to state such a fact before an engineering society. However, the usual form of such a double Pitot tube is with the openings in the tube 180 degrees apart. In fact, according to the theory of the Pitot tube, such would be the proper arrangement of tubes for a differential velocity gauge. However, since Fig. 62 represents approximately the state of affairs, this theory is incorrect, and the double or single Pitot tube cannot be used as a primary velocity gauge, though, of course, it can be used as a secondary instrument by suitable calibration.

To show the necessity of 90 degrees divergence of openings to give accurate results (in particular with regard to the direction of flow of air, which here was highly important), I explained, at some length, the apparatus used and the experiments leading to its adoption. With regard to Professor Carpenter's statement about the exceedingly crudely constructed gauge I used, I will state that in order to get point measurements of velocity it is necessary to have such a form of gauge without extended ends, and that though the gauge was crudely constructed, the fact that I was enabled to read the directions of air within one-half degree with ease made the instrument amply satisfactory from a practical standpoint, and brought out very clearly what I was desirous of ascertaining, namely, the radial velocity distribution of air at the periphery and at entrance to the fans.

With regard to Pitot tube measurements, I might mention one case which seemed rather strange at first sight. I was conducting some experiments using a box A, into which air was forced under pressure from a centrifugal fan. In the front end of the box was an opening consisting of a short piece of pipe P, with a flange F soldered to it. I took a piece of sheet iron, S, much larger than F, and held it up against the flange F. To my surprise, when but a short distance from F, it was sucked up to it quite strongly. As there was a pressure of $1\frac{1}{2}$ ounces in the box

at the time, I was at a loss to account for such a thing. Finally, I discovered that F had been loosened at the back from P, allowing a small leak of air between to shoot out radially. I next put a Pitot tube in the space B, and obtained a negative reading. However, on turning the tube in all directions, the reading remained the same, showing a partial vacuum and no flow of air. This was caused by the leak, though it was so small that the vacuum hardly seemed probable.

In another experiment, I made measurements on the flow of air around certain short curves, attempting to use a single Pitot tube for ascertaining the velocity. Of course, the velocity was highest at the outside of the curve, and gradually diminished toward the inside. The readings of the gauge connected to the Pitot tube went from a positive value at the outside of the curve

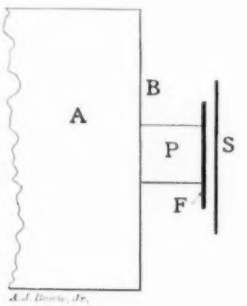


FIG. 66A.

gradually to a negative value toward the inside, the lowest negative value remaining constant over a certain area. This apparently indicated a flow of air where the negative values occurred. However, on revolving the tube, the negative values remained constant, showing that there was no flow of air. Hence, it is well to use a little caution in jumping at conclusions on the results of single readings of Pitot tubes.

I have designed fans, as nearly as possible, in conformity with the six fundamental laws laid down in the paper. Part of the design was determined theoretically, and part from practical tests. The efficiency tests of these fans have been so satisfactory as to justify, beyond a doubt, the fact that my conclusions were correct. In comparison with fans on the market here, the efficiency was very much higher, and one result of this was that, though

the fans had the same size suction, same diameter, same angle of vanes at periphery, and a smaller width of fan at periphery, still, owing to the decreased losses, the output of the fan was considerably higher than that of other makes of fans on the market here, when run at the same speed.

The same general principles of design brought out in my paper will apply equally well to fans of the Sirocco type.

No. 1064.*

THE BURSTING OF FOUR-FOOT FLY-WHEELS.†

BY CHARLES H. BENJAMIN, CLEVELAND, O.

(Member of the Society.)

1. Experiments made hitherto by the writer and described in former papers have been confined to the testing of wheels, whose diameters were two feet and less. The first wheels experimented upon were only fifteen inches in diameter and could be regarded as models, nothing more. Enough was learned from these tests to make further investigation desirable, and in succeeding years wheels twenty-four inches in diameter, of various types of construction and of various materials, cast-iron, steel and wood, were tested to destruction.

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For previous discussions on this topic consult *Transactions* as follows:

No. 85, vol. iii.: "The Fly-wheel." William Johnson. (See *American Machinist*, May 13, 1882.)

No. 497, vol. xiii., p. 618: "A Novel Fly-wheel." Charles H. Manning.

No. 515, vol. xiv., p. 251: "Strains in the Rims of Fly-band-wheels Produced by Centrifugal Force." J. B. Stanwood.

No. 565, vol. xv., p. 147: "Strength of Rim Joints in Fly-band-wheels." J. B. Stanwood.

No. 621, vol. xvi., p. 208: "Stresses in the Rims and Rim Joints of Pulleys and Fly-wheels." Gaetano Lanza.

No. 796, vol. xx., p. 125: "Note on the Strength of Wheel Rims." A. K. Mansfield.

No. 800, vol. xx., p. 209: "The Bursting of Small Cast-iron Fly-wheels." C. H. Benjamin.

No. 823, vol. xx., p. 944: "Rolling-mill Fly-wheels." John Fritz.

No. 835, vol. xxi., p. 262: "A Broken Fly-wheel and How it Was Repaired." James McBride.

No. 859, vol. xxi., p. 317: "A Note on Fly-wheel Design." A. J. Frith.

No. 907, vol. xxi., p. 955: "Determination of Fly-wheels to Keep the Angular Variation of an Engine Within a Fixed Limit." J. I. Astrom.

No. 921, vol. xxiii., p. 168: "Bursting of Small Cast iron Fly-wheels." C. H. Benjamin.

2. The interest taken in these experiments seemed to warrant carrying them further on a larger scale, such as would correspond with sizes actually in use. Four feet was selected as a limiting diameter and a cylindrical steel casing was built having an internal diameter of five feet. Figs. 67 and 68 show the exterior and interior appearance of the casing when completed. The rim or shell was made of steel $1\frac{1}{4}$ inches thick, having a tensile strength of about 65,000 pounds per square inch. The upper and lower

FIG. 67.—THE CASING IN POSITION.

halves of the rim were flanged at the junction and bolted together by one-inch steel bolts. The sides were of steel $\frac{3}{8}$ inches thick, dowelled to the rim and secured by through bolts outside and inside the rim as may be seen from the illustrations.

3. It was not deemed safe to conduct these experiments inside a building, and subsequent developments proved that it would not have been safe, for the building at least. The casing was therefore located just outside the building where the tests were to be made, in a pit or excavation lined with brick, as is shown in Fig. 67. The flanges of the lower half rested on brick piers and were bolted in place. The entire upper half of the casing could be hoisted up, giving access to the interior for setting or removing the wheels.

4. The shaft carrying the wheel to be tested turned in bearings bolted to angle irons on the lower halves of the side plates, and was connected to the driving mechanism just inside the building by a flexible sleeve coupling.

5. After the wheel was in place the casing was lined with wooden blocks to absorb the momentum of the flying fragments, as illustrated in Fig. 68.

6. Instead of using a steam turbine as in former experiments, the fly-wheel shaft was speeded up by means of a Reeve's variable

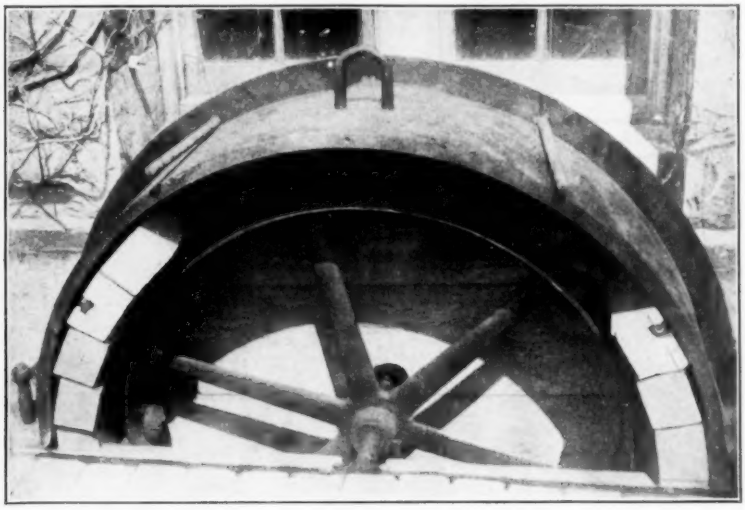

FIG. 68.—THE CASING OPEN, WITH WHEEL NO. 1 IN POSITION.

speed counter-shaft, interposed between line-shaft and the driving-shaft.

7. The first wheel to be experimented on was a well-proportioned cast-iron pulley, such as is used on shafting for transmitting power. This pulley No. 1 is shown in position inside the casing in Fig. 68. It was 48 inches in diameter, had six arms and weighed 194 pounds. The rim was whole and was $8\frac{1}{2}$ inches wide and about $\frac{3}{8}$ inches thick, finished on the outside. The arms were elliptical in section, $3\frac{1}{8}$ inches by $1\frac{1}{16}$ inches at the hub, and 2 inches by $\frac{3}{4}$ inch at the rim. On the whole the wheel was well-designed and showed no signs of shrinkage strains. It had, however, been balanced in the customary manner by riveting a cast-iron washer inside the rim at the lighter side, and this proved

its undoing. (See Fig. 68.) The combination of a thin place in the rim, a rivet hole and a heavy mass of cast-iron, is enough to wreck any wheel. As shown in the cut it was necessary to still further weight this side by winding lead wire around the arm just inside the rim. As has been shown by previous experiments on whole rim wheels of cast-iron, a bursting speed of 400 feet per second may be reasonably expected.

8. The circumference of a four-foot wheel being about 12½ feet, such a wheel should burst at about 32 revolutions per second, or

FIG. 69.—FRAGMENTS OF WHEEL NO. 1.

1,920 revolutions per minute. The pulley in question burst at 1,100 revolutions per minute, as recorded by a tachometer connected to the driving-shaft. The appearance of the wheel after breaking is shown in Fig. 69. The balance weight weighed 3½ pounds, and its center was approximately 23 inches from the axis of rotation. At 1,100 revolutions per minute the centrifugal force of the balance weight alone would be 2,760 pounds.

9. Add this radial pressure at a weak point between the arms to that due to the weight of the rim itself, and the low bursting speed is easily accounted for.

10. The linear speed of the rim at rupture was 230 feet per second. As 100 feet per second is considered the limit for belt speed, this pulley would have a working factor of safety of $(2.3)^2$

or 5.3. But suppose the rim had been a little thinner and consequently a bigger weight had been put on with a larger rivet?

11. Wheel No. 2 is shown after rupture in Fig. 70. It was a cast-iron pulley of the same general style and dimensions as No. 1, but with a split hub and rim as may be seen from the cut. The balance-weight was present here as in the former case, but was obliged to yield the palm to its rival, the flanged joint. The wheel had been cast in one piece, as is usual in such cases, with cavities cored at the joints of rim and hub.

FIG. 70.—WHEEL NO. 2 AFTER BREAKING.

12. After finishing it had been broken apart by wedges, making a fracture joint. The flanges, being located midway between the arms and bolted at some little distance inside the rim, were in the worst possible position to withstand the bending action due to centrifugal force, and their own weight only aggravated the difficulty.

13. The flanges shown in the foreground of the illustration weighed with their bolts 7½ pounds. This wheel burst at less than 700 revolutions per minute, the tachometer not recording below this speed. The writer believes the speed to have been about 600 revolutions per minute. At this speed the centrifugal force of the flanges on one side would have been 1,680 pounds. At 600 revolutions per minute the linear speed of rim would be only 125

feet per second. At the very common belt speed of 4,500 feet per minute the factor of safety would have but 2.8, which is altogether too low, considering the nature of the material and the shocks to which a pulley may be exposed.

14. There is one instance on record of the wrecking of an engine by the breaking of a generator pulley, which had a heavy balance weight inside the rim.

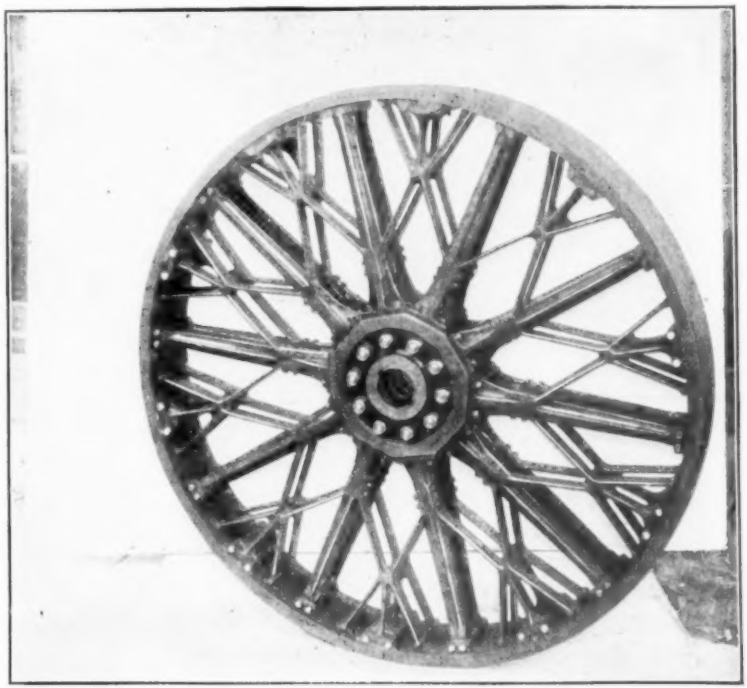

FIG. 71.—WHEEL NO. 3 BEFORE TESTING.

15. It was reserved for wheel No. 3, to develop the most dramatic series of incidents of any yet experimented upon, big or little. Fig. 71 is from a photograph of this wheel taken just before setting in the case.

16. It measured 49 inches in external diameter and weighed about 900 pounds. The rim was $6\frac{3}{4}$ inches wide and $1\frac{1}{8}$ inches thick, and was built of ten segments, the material being cast-steel. Each joint was secured by three prisoners of an I-section on the outside face, by link prisoners on each edge, and by a dove-

tailed bronze clamp on the inside, fitting over lugs on the rim. The arms were of phosphor bronze, twenty in number, ten on each side and were a cross in section.

17. These arms came midway between the rim joints and were bolted to plane faces on the polygonal hub. As shown in the cut the rim was further reinforced by a system of diagonal bracing, each section of the rim being supported at five points on each side, in such a way as to relieve it almost entirely from bending.

18. The braces like the arms were of phosphor bronze, and all bolts and connecting links of steel.

19. This wheel was designed by a Baltimore firm as a model of a proposed 30-foot flywheel.

20. On account of the excessive air resistance it was found necessary to enclose the wheel at the sides between sheet-metal discs, before any great speed could be attained. Even then repeated trials failed to reach a speed of more than 800 or 900 revolutions per minute on account of the great inertia of the wheel, and the consequent slipping of belts. By putting on more and wider belts, by a liberal use of cling-surface and with the aid of a 7½ horse-power electric motor belted on in parallel, it was found possible to get a speed of 1,650 revolutions per minute.

21. After the wheel had been run at this speed it was stopped and examined.

22. The inspection showed fracture of several of the I-shaped prisoners on the outer surface of the joints and a slight opening of the joints themselves, to the extent of perhaps one or two hundredths of an inch. On June 2, 1903, the casing was closed for the last time and the combination of driving mechanisms set to work. The observers were all well protected by the thick piers of the building, while other spectators were kept at a safe distance and well away from the plane of rotation. Two of the observers watched the pointer of the tachometer through opera-glasses, another kept the time, while a fourth manipulated the driving levers.

23. As the hand of the speed counter reached and slowly passed the 1,600 mark, the feeling of suspense on the part of those watching reached the acute stage. The pointer crept slowly on and as it quivered on the mark of 1,775, there was a sudden crash, a sound of rending and tearing, and the writer saw the counter-shaft inside writhing on the floor like a wounded snake. On stepping outside he was saluted by a shower of falling splinters and fine debris,

and was surprised—putting it mildly—to note the disappearance of the greater part of casing and wheel.

24. Fig. 72 is from a photograph taken immediately after the explosion and shows the completeness of the wreck.

25. The steel rim of the casing was broken off short six inches below one of the flanges, and the entire upper half weighing half a ton was projected about 75 feet into the air and landed some hundred feet away on the campus. See Fig. 73.

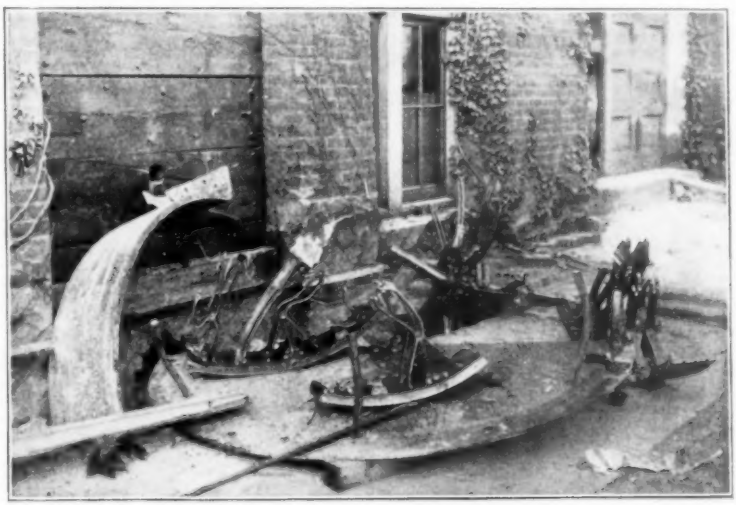

FIG. 72.—AFTER THE EXPLOSION.

26. On its way up it carried away part of the cornice of the building, and this collision was probably what caused it to deviate so much from a vertical path.

27. The hub and main spokes of the wheel remained nearly in situ, but parts of the rim were found two hundred feet away, while one large fragment landed on the roof of the building.

28. This sudden failure of the rim casing was unexpected, as it was thought the flange bolts were the parts to give way first. The tensile strength of the rim at the point of fracture was about 1,200,000 pounds, or about four times the strength of the wheel rim at a solid section. Examination of the break in the casing showed a clean, bright fracture, with almost no imperfections.

29. The failure of the wheel itself was due to a gradual opening of the joints, occasioned by the fracture of the outside pris-

oners and to flaws in the bronze castings of the arms near their junction with the rim.

30. On putting the pieces of the wheel together in their original order it was easy to locate the joint which first gave way, on account of the symmetry of the breaks either side of a diameter through this point. It is but fair to the builders of the wheel to say that the fractures showed uniformity of strength and of workmanship, since there was hardly a member or a joint which did not fail in one part or another of the wheel.

FIG. 73.—FRAGMENTS OF CASING.

31. One thousand seven hundred and seventy-five revolutions per minute means a linear speed of rim of 22,300 feet per minute, or 372 feet per second. This is not as great as the probable speed of a solid cast-iron rim of good design, but it is greater than the speed of any sectional or jointed rim, which has been tested.

32. The tensile stress due to the centrifugal force at this speed is 13,800 pounds per square inch. This shows that the joints were much weaker than the solid rim.

33. On the whole the test of this particular wheel was disappointing, since its strength was not sufficient to repay one for the expense of the design. Fig. 74 illustrates a model wheel of cast-steel which has not yet been tested.

34. The apparatus for the experiments above described was designed and put in position by Messrs. Moulton and Wachalofsky, members of the class of 1903, Case School of Applied Science, and the tests on wheels Nos. 1 and 2 were made by them. For the experiments on the last wheel credit should be given to Mr. Geo. E. Daniels, instructor in mechanical engineering, whose

FIG. 74.—CAST-STEEL MODEL WHEEL.

patience and ingenuity overcame all the obstacles to obtaining high speed with so heavy a wheel.

35. It is interesting to compare the kinetic energy of the rim of the wheel at the recorded speed with the work of destruction. Assuming the rim with its lugs, flanges, etc., to weigh 300 pounds, which is a reasonable estimate, the kinetic energy at a speed of 372 feet per second would be 645,000 foot pounds.

36. Further assuming that none of the energy was dissipated in heat, and that the combined mass of wheel and casing projected into the air weighed 1,500 pounds, we find the height of

projection to be 430 feet. It is good cause for congratulation that four-fifths of the energy *was* dissipated.

37. The experiments which have just been described make it clear that more than ordinary precautions must be taken to insure safety. The method adopted by an engineer in Germany for testing emery wheels, seems to the writer well adapted for this purpose.

38. It consists in mounting the wheel to be tested on the lower end of a vertical shaft and sinking the apparatus in a hole in the ground.

39. The writer hopes to be able to make some experiments on large wheels during the coming year. A pit about five feet in diameter by three or four feet deep will be excavated in a gravelly soil, sheet piling being used to enclose the sides. The wheel will be fastened to the lower end of a vertical shaft, running on specially designed thrust bearings and driven by a friction speed controller at the top.

40. A steam turbine will be used as a motive power, since this is easier to regulate and less susceptible to injury than the electric motor.

41. Acting on a recent suggestion the writer also hopes to test grindstones and emery wheels in a similar manner.

DISCUSSION.

Mr. H. H. Suplee.—I would like to say that the method referred to on the last page of this paper, of testing the wheels on a vertical shaft was described very fully in the proceedings of the Society of German Engineers for 1902. Professor Grübler cites tests of a great many emery wheels in that way. He had a deep shaft constructed and the recording apparatus at the top was very easy to examine and it kept in perfect condition up to the very moment of bursting because of the entire safety of the operation. A full account of these trials, together with figures of the broken portions of the wheels and the record of the whole set of tests on emery wheels may be readily found by those interested in the subject.

Mr. Gus C. Henning.—I would like to call attention to the great advance that has been made in this line. We are getting every day through these tests more information, which will be

of great value to us. It goes to show that certain materials are superior to others, and I think that this calls attention to the fact that fly-wheels should be made with steel rims, because the steel rim can be welded so thoroughly that it is equally strong at all sections and accurately balanced. It shows too that in many cases it would be more economical to use steel rim wheels than cast iron wheels.

Mr. Geo. R. Stetson.—I have had some disagreeable experience in the bursting of wheels in use. The method of putting the balance weight on is certainly objectionable. As it is now it is frequently set on the wheel with a limited base area, and if centrifugal force acts it has a tendency to throw it through the rim of the wheel. Another thing that I have thought might contribute to the breaking of larger wheels is the heating of the outside of the wheel by the slipping of a belt, the engineer perhaps not being conscious of the fact that the wheel was running through the belt. I have known a large engine wheel to have quite a degree of heat that I should think would lead to the expansion of the metal in the wheel, and it was without the observation of those that were in attendance. There was a very disastrous break of the wheel in the Amoskeag Works some years ago, as perhaps you will remember. Mr. Manning ultimately built up a wheel, making the rim of wood, so that it would obviate the trouble, area expansion, and I think he has entirely done so. I think it would be quite possible to carry on a line of experiments to know if there was anything in that cause. I know in one case where the wheel ran through the belt until it became quite disagreeable to put your hand on the outside of the wheel. It was an eighteen foot wheel attached to a 300 horse-power engine. The heat was enough to expand the metal. I think experiments could be made which would prove this theory. The worst trouble that we meet is in the casting, where the shrinkage is not properly taken care of and the spokes are drawn out from the rim. By greater skill in casting that has been obviated, but in my experience in electric light stations we have thrown around iron enough to bring about the subjugation of Port Arthur if it had been properly directed.

Mr. J. E. Johnson, Jr.—I would like to suggest that a wheel built of segments with joints half way between the arms is essentially a weak construction, because no matter how strong the joint may be in tension, it is necessarily very weak in transverse

strength unless made so deep as to be impracticable, and as the transverse strength is as important as tensile strength under the action of centrifugal force, wheels so made are inherently weak.

For high-speed wheels expense does not seem to be a controlling consideration as shown by the model wheel with diagonal braces shown in this paper which must have been very expensive, and a rational solution of the problem under such circumstances would seem to be afforded by making the joints in the center of the arms that is making each section consist of two half arms with a whole section of rim between them with the faces of the half arms planed to fit each other, at the ends only, or throughout their length as might be thought advisable.

The joint could be made stronger than at present with the same metal and the individual sections would be approximately triangles, the strongest and stiffest of all shapes, instead of a single tension piece with two weak projecting cantilevers, which is one of the weakest.

The expense of fitting would probably be somewhat more than that of the present method, but not necessarily very much, and the weight with good design would be less.

As Professor Benjamin is doing this valuable work in elucidating this subject I would like to suggest this construction for his consideration and I believe it would be a matter of deep interest to this Society if he would build and test such a wheel in comparison with a similar one of the ordinary type.

Mr. F. A. Halsey.—I would like to suggest the desirability of testing the wheel that Mr. Fritz described before the Society some years ago, and which has always seemed to me a very remarkable piece of work. I believe it is possible to design a wheel of that type with a rim joint efficiency of 100 per cent., a result that is not possible with any other type that we know of. It seems to me that this wheel is entitled to more attention than it has yet received.

*Prof. C. H. Benjamin.**—The greater cost of steel wheels will hinder their general adoption by engine builders although several such have been designed and constructed. The writer would refer to an article on "The Development of Fly-Wheels" in Cassier's Magazine for July, 1900, where several steel built-up wheels are illustrated and described. There is no question but such a wheel

* Author's Closure, under the Rules.

can be built with a rim efficiency of 90 or 95 per cent. and a bursting speed more than double that of any cast-iron wheel.

For the present, however, the problem is how to build a cast-iron wheel which shall be safe at any speed likely to be used in engine practice.

The wheels built by Mr. John Fritz and alluded to by Mr. Halsey are good examples, but they are not suitable for belt wheels on account of the narrow rims.*

Previous experiments by the writer have shown that the joints in the rim of the rolling mill or blowing engine type of wheel can easily be made to have a high efficiency, while the wide, thin rim of the belt wheels presents a much more difficult problem. Mr. Johnson's remarks are right to the point. The solution for this class of wheel is to have the joints over the arms and tests on such joints reported in a previous paper * show them to be much stronger than joints between the arms.

It is interesting to note the progress that has been made in this regard since this Society first began to discuss the subject, as indicated by the footnotes at the beginning of this paper. In 1893 Mr. Stanwood called attention to the bending of the rim between the arms and followed this up the next year by a discussion of rim joints. In 1895 Professor Lanza presented a mathematical discussion showing the probable extent of the transverse stresses. Nearly every year has seen some contribution to the subject, either by experiment or discussion.

Ten years ago nearly all engine builders made fly-wheels with flanged joints mid-way between the arms; now, few of them would venture to do so. The writer could name several representative makers of Corliss engines who have made a radical change in this respect.

A very good example of the right way to make a joint in a wide wheel-rim is illustrated in Cassier's Magazine for July, 1900, page 253, Fig. 8.

The writer expects to construct a wheel pit the coming year for testing wheels to destruction, and would be glad of contributions in the shape of model wheels not to exceed four feet in diameter.

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* *Transactions*, vol. 23, p. 175.

No. 1065.*

AN INDICATING STEAM METER.

BY C. E. SARGENT, CHICAGO, ILL.
(Member of the Society.)

1. That there has been for many years and is an ever increasing demand for some practical device which will measure or indicate the amount of steam in pounds which is delivered through pipes to an engine, radiator, or steam pump is apparent to the engineer as well as the manufacturer and the tenants and landlord who buy or sell steam for power or for heat.

2. A simple, cheap and fairly accurate device, applied to the supply pipes of high duty engines and low duty direct acting steam pumps should do more toward revolutionizing the present methods of feeding boilers, pumping water for elevators and kindred purposes than all the testimony of mechanical engineers as to the inefficiency of such means. Seeing is believing, and when the manager can see that a certain steam pump is taking six or seven times as much steam per horse-power hour as the automatic engine by its side, then power pumps and more efficient apparatus will be put in. The expense of making efficiency tests of small plants is so great, that probably not one engine in twenty is tested for economy, and the salesman with the inefficient engine, knowing that there is but a slight probability of any test being made, makes as good a guarantee as the competitor with a more efficient machine. If a steam meter in the supply pipe would show exactly the amount of steam going into the engine, any leak or accidental derangement of the valve setting would be instantly discovered with a constant or regular load on the engine.

3. The need of a steam meter has been so great that several devices have been designed for the purpose, but on account of the

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

changing conditions of the product with which we have to deal, only partial success has been attained.

References: U. S. Patent 729511,
U. S. Patent 481287,
Transactions A. S. M. E., 1903,
The St. John Meter, New York City.

4. If we were dealing with a permanent gas of a practically uniform pressure and temperature such as the illuminating gas used for heat and light, then the metering of steam would be a simple problem, but steam is not a perfect gas and usually varies in pressure and temperature many pounds and degrees, and contains more or less water depending on the distance from the boiler, and the efficiency of the insulation of the radiating surfaces.

5. As the steam pressure in various plants varies from below atmospheric pressure when used for heating to as high as 200 pounds per square inch where used for power in multiple cylinder engines and steam turbines, meters with considerable range are required, no matter where or for what purpose they are to be used; and to design a meter to operate successfully under these varying conditions has been the aim of the writer, and the device herein described is believed to overcome many of the difficulties which have heretofore been obstacles to success.

6. If there is a constant difference between the pressures each side of a certain opening, the amount of steam flowing through that opening will depend on the area and the density of the steam which varies with the pressure. If the initial pressure and the difference in pressures remains constant, then the weight of steam flowing through will depend on the area of the opening and if the opening is constant, the weight of steam flowing through will vary with the pressure and the difference of pressure on either side of the opening.

7. If the difference in pressure between two tanks is 1 pound, and the initial pressure varies from 50 to 100 pounds, twice as much steam will flow through a certain opening with 100 pounds pressure as would flow through with 50 pounds pressure. In the meter under consideration, which is designed for steam pressures of 50 to 100 pounds, a drop of 2 per cent. is thought advisable, and the areas are proportioned accordingly. When steam is flowing through, the opening will adjust itself, so that the difference of pressure is always a percentage of the pressure on the inlet side.

8. Any tendency of the pressure to equalize will immediately close the valve, and by thus throttling the passage, reduce the outlet pressure below the inlet pressure as designed. Any tendency for the outlet pressure to reduce more than 2 per cent. below the inlet pressure will raise the valve and increase the area

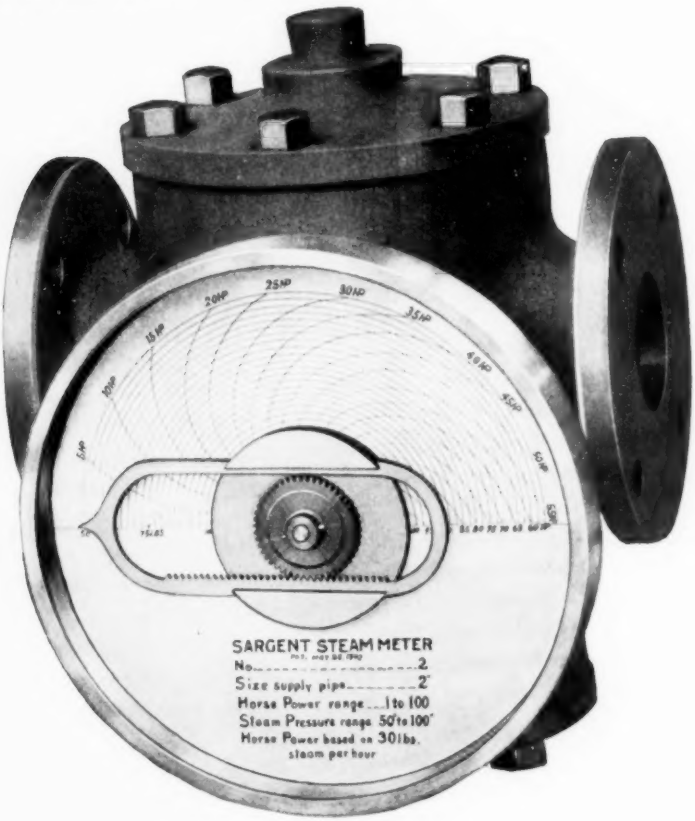

SARGENT STEAM METER
No. 2
Size supply pipe 2"
Horse Power range 1 to 100
Steam Pressure range 50 to 100
Horse Power based on 30 lbs.
steam per hour

FIG. 75

of the steam passage. Slugs of water coming with the steam cannot by inertia open the valve D because they cannot strike the valve except radially, and it is balanced against such a force. The inertia of water striking unprotected floating valves or vanes of steam meters at 100 feet or more per second, would necessarily cause derangement. As the opening F is small and changes the direction of any water which might tend to go through should

the valve suddenly open, no harm can arise from water in steam, except, possibly, to introduce an error in indication.

9. We see now that the main valve stem E will stand between the position of the full opening or the complete closure of the valve, depending—of course—on the amount of steam passing through, and the position of this valve stem determines the movement of the indicating finger around the center of the dial through mechanism hereafter described, and shown in Figs. 75, 77 and 78.

10. If the steam always had the same density, that is—if a certain volume always had a certain weight, no further indicating apparatus would be necessary, but as the weight per volume varies with the pressure, the same quantity of steam under double the pressure will flow through one-half the area for the same length of time; also if the difference in pressure increases with the absolute pressure, more steam will flow through a certain opening when the pressure increases than if the difference in pressure each side of the opening remained constant.

11. On the assumption then, that pressure times volume equals a constant ($PV = C$) which is near enough for practical purposes, we can determine the amount of steam passing from one receptacle to another, if we know the difference in pressure in the two receptacles, by determining the area of the opening through which the steam flows, but as the weight of steam passing through a defined opening will vary with the pressure, and the difference in pressure each side of the opening, then we must vary our opening with the pressure if a uniform weight is flowing through. And this is the principle upon which this meter is based. With a constant weight or horse-power passing through, the pointer will follow the horse-power curve on the dial though the pressure of the steam varies between the limits of the meter.

12. Meters having an adjustable opening through which the steam flows have been made and with a constant pressure record fairly accurately the pounds of steam passing through them, but with a variable pressure, from the nature of the device, the indication would be inaccurate.

Then again, in any steam pipe in which the steam is not superheated, there is more or less condensation, which, as the demand for steam is suddenly increased, is carried along in the shape of slugs or charges of water, the inertia of which, travelling at the rate of 100 or more feet per second, will seriously impair the accuracy of a meter susceptible to derangement by impact. While

any steam meter should have a separator between it and the boiler, water can in no way affect the mechanism of the meter herein described, as all moving parts are protected from derangement caused by the inertia of water passing through the openings.

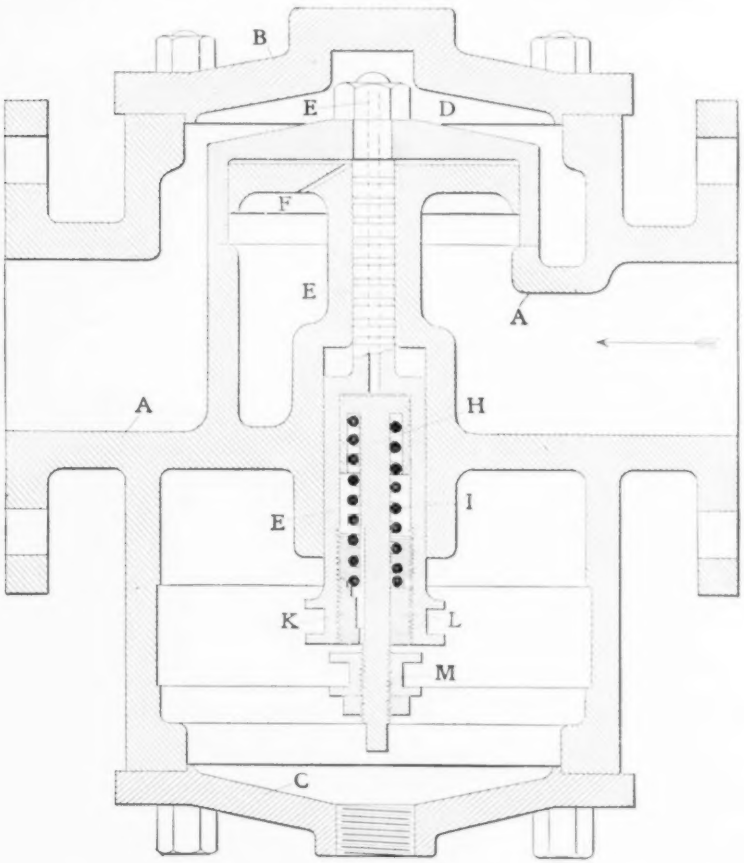


FIG. 76

Description of Meter.

13. Fig. 75 is a half-tone of a 2-inch meter showing the dial side and compensating pointer. It is flanged and faced and can be inserted in the steam pipe as an ordinary 2-inch globe valve. This particular meter is calibrated to show the horse-power (based

on 30 pounds of steam per horse-power hour) passing through with a steam pressure of between 50 and 100 pounds.

Fig. 76 is a longitudinal section.

Fig. 77 a transverse section.

Fig. 78 an elevation of the indicating mechanism looking towards the back of dial.

14. Referring to Fig. 75, if no steam is flowing through the meter, the pointer will indicate zero horse-power as in the figure, but will indicate the pressure if it ranges between 50 and 100 pounds. As the end of the pointer will not begin to move towards the center of the dial until the pressure reaches 50 pounds, it will stand at 50 pounds, when there is no pressure in the meter.

15. If the pressure were maintained exactly at 50 pounds and 150 pounds avoirdupois of steam passed through per hour (5 horse-power at 30 pounds of steam per horse-power hour) then the pointer would stand at 5 horse-power, and 50 pounds pressure.

If 40 horse-power were passing through, the pointer would so indicate, and if the pressure were raised from 50 to 100 pounds, the end of the pointer would follow the curve of 40 horse-power from the 50 to the 100 pressure limit.

In like manner the pointer would indicate the weight of steam passing through, no matter how the quantity or pressure varies, provided it remains within the limits of the machine.

In Fig. 76, which shows the steam passage and moving parts, *A* is the body of meter, *B* the top cover, and *C* the bottom cover; *D* is the self-adjusting valve which remains seated when no steam is passing through the meter, and which is held in place by and fastened to the valve stem *E* which works in two diameters in the body *A*.

The bottom cover *C* is tapped for $\frac{3}{4}$-inch pipe which connects the meter to the atmosphere or sewer.

16. The action of the meter is as follows: steam is admitted on the side of the arrow and surrounds the valve stem guide; as soon as pressure accumulates it passes through the small hole *F*, and raises the self-adjusting valve *D*, allowing the steam to pass to the outside and top of the valve.

17. As the bottom end of the stem *E* is open to the atmosphere, the pressure on top of the valve *D* tends to close same by an amount of pressure equal to the pressure in the meter into the area of the valve stem immediately below the valve. As the pressure under the valve is always tending to open same, and the

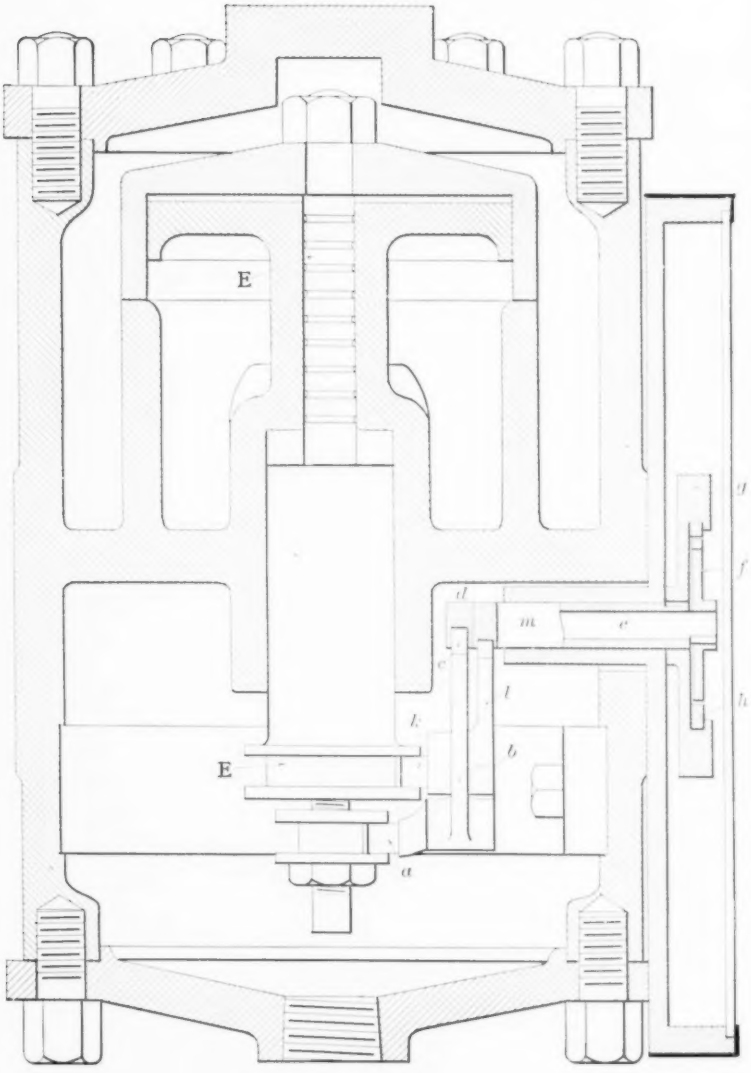


FIG. 77

pressure on the stem is always tending to close same, there will be a difference in pressure between the inlet and outlet of the meter equal to the difference in area of the valve and valve stem.

19. As these areas are so designed that the ratio is about 50 to 1, the pressure of steam on the discharge side of the meter will be

2 per cent. less than on the admission side with any pressure carried.

If 100 pounds pressure is carried on the boiler side of meter 98 pounds pressure will be delivered on the engine side and if the admission pressure is only 50 pounds the discharge pressure will be 49 pounds.

21. By a proper proportioning of the valve and the valve stem areas, a minimum drop of pressure of the steam passing through, practically, with the pressure per square inch, we see that some provision must be made for a change of pressure in the steam passing through the meter.

22. Referring to Fig. 76, the pressure of steam above the valve *D*, and on the outlet side of the meter, may pass through the hollow valve stem *E*, and act on the piston *H*, compressing same, and the spring *I*, which—in the present case is of such a tension that 50 pounds pressure per square inch is necessary to overcome the statical conditions of same, and 100 pounds per square inch is sufficient to compress it to its full limit.

23. The movement of this piston *H*, is transmitted to the roller *A*, Figs. 77 and 78 which brought the bell crank *b*, the segment *c*, the pinon *d*, the shaft *e*, and the large pinion *f*, Figs. 75 and 77, and the rack *h*, transmits the movement of the end of the pointer to and from the center of the dial, showing the steam pressure per square inch on the meter.

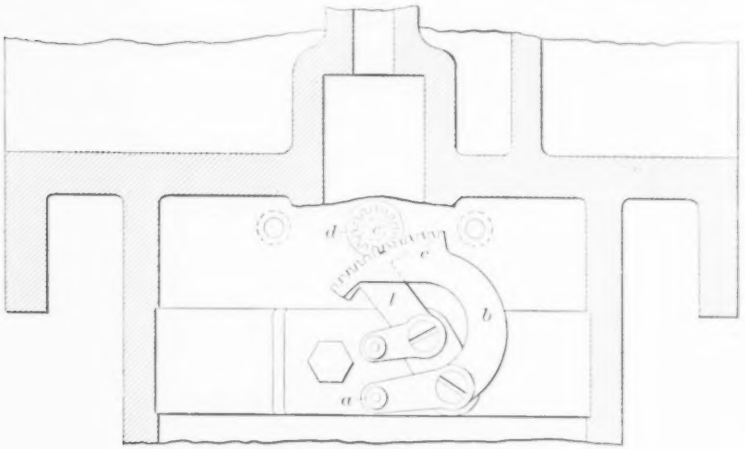
24. The position of the main valve stem *E*, is transmitted through the roller *k*, the bell crank *l*, the hollow shaft and pinion *m*, and the disk *n*, which revolves the pointer around the dial center. The amount of movement around the center depends upon the opening of the valve, and the distance of the end of the pointer from the center of dial depends on the steam pressure per square inch between the limits of the meter of 50 and 100 pounds, and, on account of the compensating mechanism, the pounds avoirdupois of steam or horse-power passing through is always indicated.

25. While water in the steam cannot derange the action of the meter, it is advisable to place a separator between meter and boiler as near the meter as possible, in order to insure accuracy of indication.

26. The size of the pipe and meter determines the maximum possible horse-power on the dial, while the number of the spring determines the range of pressure required.

27. In measuring exhaust steam for heating, the spring and dial are designed to show a pressure from 0 to 30 pounds absolute, and the dial may indicate pounds of steam instead of horse-power.

If the steam pressure upon which the horse-power of the engine is based is 150 pounds then the meter spring and dial would show a range of horse-power from 0 to full meter capacity and a pressure range from 125 to 175 pounds or greater range if desired.



Sargent, C.E. SECTION OF PLAN AND ELEVATION OF INDICATING LEVERS. Am. Bk. Note Co., N.Y.

FIG. 78

28. By carrying extra springs and dials to correspond, meters for any range may be furnished from stock on short notice, fully graduated for the pressure desired.

It was the intention to calibrate each meter sent out, but when two or more of the same size are calibrated in series the indications correspond so closely that it is expected that the calibration of only one of a size and range will be found sufficient.

29. The method of calibrating a meter is as follows:

From the meter's construction there must always be a difference in the steam pressure each side of the meter depending on the inlet pressure; therefore, the amount of steam flowing through will regulate the amount of valve opening.

As the difference in pressure between the two sides of the meter increases as the pressure increases, the opening through which the steam flows will not increase as fast as the pressure, in order to

let the same weight of steam through in a certain time, and this is advantageous, because the valve will require less movement as the pressure increases.

30. In order to calibrate the meter, it is necessary first to get a spring of the proper tension to move the end of the pointer from the outside to the inside of the dial through the range of pressure required. The spring is first calculated, then can be adjusted when hot and in place, if necessary, by holding the outside valve stem *E*, and turning the inside valve stem with a socket wrench.

31. The meter is connected up between the steam header and condenser with a pressure regulating valve in series between the meter and steam supply.

Accurate gauges are connected to both the inlet and outlet side of the meter, and the pressure regulator is set to carry the highest pressure for which the meter is calibrated. A valve on the discharge side is opened slightly, and the position of the pointer noted, and the quantity of steam flowing through is condensed and weighed. In like manner a series of points for a constant pressure is established through the whole capacity of the meter. The pressure is reduced and the operation is repeated. When completed and the points are connected, there will be a series of concentric rings representing the different pressures, and a series of converging or diverging curves representing the pounds of water passing through which is indicated by the position of the pointer when meter is in operation.

32. The spring should be of steel, heavily nickel-plated. This meter may be inserted in the steam pipe, either next to the boiler, in which case no separator need be used, or next to the engine, in which case a separator between the meter and boiler as close to meter as possible should be used, and a drum having four times the capacity of the first cylinder at its average cut off should be placed between meter and engine to get a constant flow through the meter and a practically stationary position (non-oscillating) of the pointer. By observing the position of the pointer for any interval of time, the amount of horse-power or pounds of steam passing through to the engine or heating system may be determined.

DISCUSSION.

Mr. Gus C. Henning.—I think Mr. Sargent has presented to us a very interesting and ingenious device for doing the work for which it is intended. Of course it is a steam mechanism, and

as such we will undoubtedly find that on account of the various surfaces in contact and in motion at times and at others quiescent there will be corrosion, and I am afraid unless the apparatus is carefully examined from time to time and kept well lubricated it will meet the fate of many another steam apparatus and stick, thereby becoming inoperative. The increased friction between corroded surfaces will also cause inaccurate indication of loads and pressures. But I think the design is really very ingenious and certainly ought to find considerable application in innumerable places where we now have no record of work done or of steam consumption, or where the coal is going. I am very sorry that Mr. Sargent is not here with us, because I think he could tell us how many of these meters have been in use, and could give us some idea how often the part *E* sticks, and then at the lower end of Figure 77 and in Figure 76, *F* and *D* and the column 3 in the center part of the body *H* and the lower part which also moves in the larger cavity in *H*.

Mr. H. H. Suplee.—I would like to call attention to paragraph 12.

"Meters having an adjustable opening through which the steam flows have been made and with a constant pressure record fairly accurately the pounds of steam passing through them, but with a variable pressure, from the nature of the device, the indication would be inaccurate."

There has been made in Germany what is called the Gehre steam meter, constructed upon the principle of measuring the flow by the difference in pressure upon the two sides of an orifice, the variation in pressure being corrected by the use of gauge templates which enable the direct rise and fall of gauge pistons, acting in direct proportion to the variations in pressure difference, to record upon a moving sheet the ordinates corresponding to the flow at every moment. This instrument, which is in practical use in Germany, is described very fully, with illustrations, in the issue of the *Zeitschrift des Vereines Deutscher Ingenieure* for December 8, 1900.

*Mr. C. E. Sargent.**—In regard to the remarks of Mr. Henning will say that several of these meters have been in use for over a year, giving in most cases very good results.

In two cases where the steam was very wet a deposit of cal-

* Author's Closure, under the Rules.

careous matter in the pressure cylinder and on the main valve stem put the meter out of commission and the removal of this deposit was necessary for its correct operation.

When the meter is in operation the slightest change in the quantity of steam passing through will cause the needle to assume a new position, and if the position of the floating needle is displaced by hand it will immediately return to its correct position. On account of this property of the instrument a sticking of the working parts is quickly noticed. Calcareous deposits from the entrailed water have affected the free action of the meter more than oxidization, though time alone will prove the reliability of the instrument.

No. 1066.*

FUEL CONSUMPTION OF LOCOMOTIVES.

BY G. R. HENDERSON, NEW YORK CITY.

(Member of the Society.)

1. The fuel bills of a railroad constitute ordinarily about 10 per cent. of the total expense of operation, or from 30 to 40 per cent. of the actual cost of running the locomotive. On important systems the gross amount of coal burned assumes a very large figure—running into millions of tons. Each engine will probably consume $5,000 worth of coal in a year on the average, so that for 1,000 locomotives the annual coal bill would approximate $5,000,000. While this is one of the largest items of expense, there is probably less actually known about it than any other account. We may know in a general way that an engine of a certain class, loaded with a definite tonnage, will haul its train in a given direction over a particular division with a consumption of so many pounds of coal per 100 ton miles, but here our knowledge stops, and if it should be asked how much is used in ascending the maximum grade, how much on the subordinate grades, and what quantity on the level, there is little likelihood of receiving a correct answer.

2. The reason for this lack of definite information is not hard to find; in fact it is quite obvious. While it is a comparatively simple matter to determine the quantity of coal used on a trip throughout a run, by means of track scales and measured supplies taken en route, it is very difficult and laborious to sub-divide it between terminal points, in the proper proportion to the work done on each of the various grades. An approximation is sometimes obtained by counting the shovelfuls thrown into the fire-box between different points, but this, of course, cannot be considered an accurate method. The ordinary exigencies of railroad

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

traffic are so many and varied that it is almost impossible to maintain fixed conditions for a length of time sufficient to determine factors of unquestioned value.

3. The same comments apply equally to the consumption of water. Even if meters be placed in the feed pipes, the inconvenient location for observation and the variable methods of working the injector by allowing different heights of water in the boiler, not to speak of leaks and wastes, militate against accurate measurements. Thanks, however, to the "Locomotive Testing Plant," we are now able to work an engine for a long period under constant conditions, and at the same time make accurate measurements of the fuel and water consumed.

4. Several years ago, when connected with the Chicago and Northwestern Railway, the author was able to make a complete set of tests with the standard heavy freight engine of that road, first upon the testing plant and afterwards in road service with a dynamometer car. Using the results of these tests as a foundation, it has been found possible to elaborate a diagram for practically any particular locomotive whose general dimensions are known, which will give at once the coal consumption per mile or per hour for various grades and speeds or train loads. This diagram is based upon theoretical as well as practical considerations, and will, it is believed, give values agreeing closely with actual conditions.

5. The construction of the diagram and the method of using it can, perhaps, be made most clear, by assuming a locomotive of certain proportions, and developing the study for this engine. We will therefore consider a consolidation locomotive or 2-8-0 type having the following general dimensions:

Diameter of cylinders	21 inches.
Stroke of piston	32 "
Diameter of drivers	56 "
Boiler pressure	200 pounds.
Grate area	40 sq. ft.
Heating surface	3,200 " "
Weight of engine and tender	150 tons.

6. The theoretical tractive force of such a locomotive will be

$$T.T.F. = \frac{P\,d^2\,s}{D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad 1$$

where P = Boiler pressure in pounds per square inch.

d = Diameter of cylinder in inches.

s = Stroke of piston in inches.

D = Diameter of drivers in inches.

When we allow for drop in steam pressure and internal resistance, we find that the available tractive force at circumference of the drivers is only 0.8 of the theoretical tractive force or,

$$A.T.F. = \frac{0.8\,P\,d^2\,s}{D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad 2$$

for simple engines, when working at slow speeds with the reverse lever in the corner notch.

7. For the engine under consideration we therefore find as follows:

$$T.T.F. = \frac{200 \times 441 \times 32}{56} = 50{,}000 \text{ lb. approx.}$$

$$A.T.F. = 0.8 \times 50{,}000 = 40{,}000 \text{ lb. approx.}$$

As the speed of the locomotive increases, however, beyond the point where the boiler can supply the complete volume of the cylinders at each stroke, an earlier cut-off must be used, and it is necessary to determine the effect of such a change. In order that this may occur at the maximum possible speed, the boiler must be worked to its full capacity, which is limited by its ability to burn fuel. From various tests it seems as if this limit might be considered as stated below, the quantities being expressed in pounds of coal per square foot of grate area per hour:

Bituminous Coal	200 pounds.
Anthracite, large sizes	100 "
Anthracite, small sizes	60 "

8. We will assume that our engine is burning Pennsylvania or Virginia semi-bituminous coal, therefore the maximum combustion will be $40 \times 200 = 8{,}000$ pounds coal per hour. We admit that this is a large amount to be handled by one man for any great length of time, but there is no doubt that it could be burnt, if supplied.

9. In order to determine the quantity of steam generated by this amount of fuel in the boiler which we have assumed, Fig. 79 is introduced. This has been compiled from various sources of information, and, it is thought, fairly represents the average practice in this country. In this figure, the ordinates give the maximum evaporation in pounds of water from and at 212 degrees Fahrenheit per square foot of heating surface per hour that can be expected under ordinary conditions, as stated above, the abscissæ denoting the ratio of heating surface to grate area. For the engine in question this will be

$$\frac{3200}{40} = 80,$$

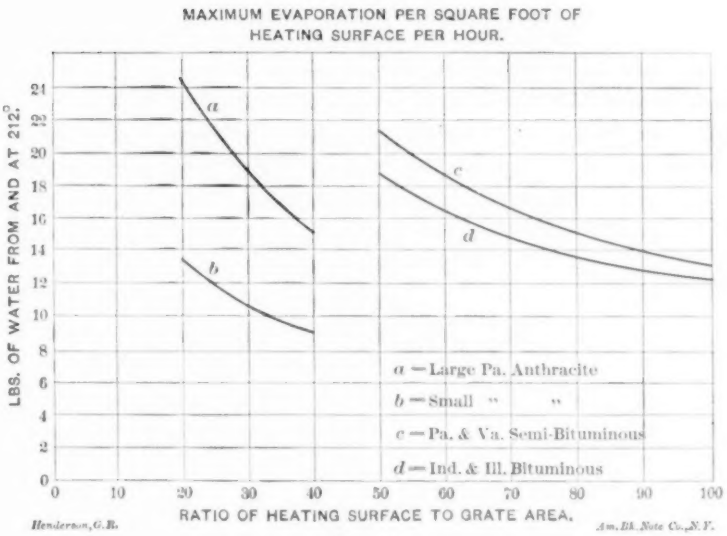


FIG. 79

and for semi-bituminous coal, curve *c*, we find that with a ratio of 80, 15 pounds of water, from and at 212 degrees, may be evaporated per hour from each square foot of heating surface, or for the boiler as a whole, $3,200 \times 15 = 48,000$ pounds per hour.

10. The factor of evaporation from ordinary temperatures of feed water, will be about 1.2, so that we shall have at boiler pressure, $\frac{48,000}{1.2} = 40,000$ pounds per hour.

11. The steam will be somewhat reduced in pressure at the cut-off point, however, and the table here given indicates the probable relation of this pressure to the boiler pressure, when the reverse lever is in the corner notch, and the throttle wide open.

RATIOS OF CUT-OFF PRESSURE TO BOILER PRESSURE:

Revolutions per Minute.	Starting.	50	100	150	200
Long Ports	.98	.94	.88	.83	.78
Short Ports	.98	.92	.85	.77	.72
Medium Ports	.98	.96	.86	.80	.75

By long ports is meant those in which the length of port in inches divided by the area of the cylinder in square inches is approximately 0.12, and by short ports, where this ratio is about

0.05. If we assume 0.90 for the ratio in the case in hand, we shall have $200 \times 0.90 = 180$ pounds at cut-off, which steam will weigh 0.432 pounds per cubic foot. The volume of a cylinder 21 inches diameter and 32 inches long is 6.4 cubic feet, or for one revolution four times this amount or 25.6 cubic feet. No allowance is made for clearance, as the cut-off, with lever in the corner, is usually about 90 or 92 per cent. of stroke. For each revolution, then, the steam consumption will be

$$25.6 \times 0.432 = 11.06 \text{ pounds}$$

and

$$\frac{40{,}000}{60 \times 11.06} = 60 \text{ revolutions per minute}$$

as the maximum speed at which the boiler will furnish steam at full stroke. The speeds in miles per hour corresponding to the revolutions per minute for a 56-inch wheel are as follows:

5	10	15	20	25	30	35	40 miles per hour.
30	60	90	120	150	180	210	240 revs. per min.

therefore it is plain at above 10 miles an hour, the cut-off must be reduced, diminishing the available tractive force of the engine.

12. A study of the variation in tractive force due to speed indicates that the method explained below gives a close approximation to actual results.

In Fig. 80 the ordinates represent the tractive force in pounds, and the abscissæ the speed in miles per hour. As the maximum speed at full stroke was found to be 60 revolutions per minute or 10 miles an hour, we find the intersection of this speed with the theoretical tractive force at A. We therefore construct an equilateral hyperbola through this point, that is a curve the product of whose ordinates will always have the same value, viz., $50{,}000 \times 10 = 500{,}000$. As we have seen, the available tractive force, however, cannot exceed 40,000 pounds. By drawing from the point B a tangent to the hyperbola, we then have a locus consisting of a straight line and a curve, and this locus gives us the maximum available tractive force (at circumference of the drivers) for which the boiler will supply the cylinders at any speed. But to do this, we must burn 8,000 pounds of coal an hour, so that the locus BC also gives the combinations of speed and available tractive force which may be obtained by the combustion of 8,000 pounds of coal an hour.

13. The rate of combustion per square foot of heating surface per hour is $\frac{8,000}{3,200} = 2.5$ and from Fig. 81 (curve c) we should expect 6 pounds of water per pound of coal from and at 212 degrees, or a total steam production of $8,000 \times 6 = 48,000$ pounds, which

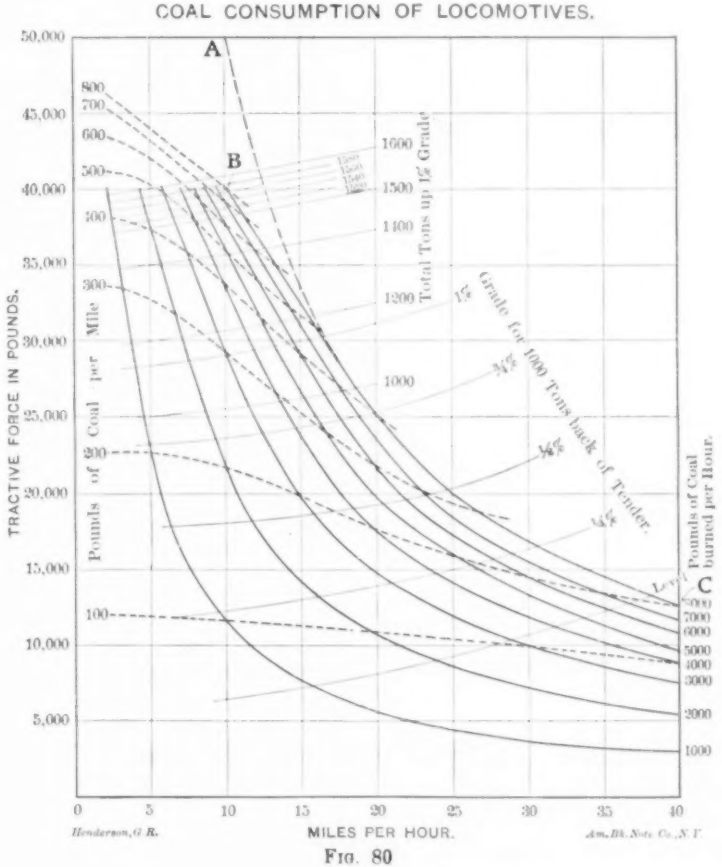


FIG. 80

is the same as our first figure. If the rate of combustion be reduced, however, there will be more steam generated per pound of coal, as indicated by Fig. 81. For instance, if $\frac{3}{4}$ the amount be consumed, or 6,000 pounds, the rate of combustion will be $\frac{6,000}{3,200}$ $= 1.87$ pounds per square foot of heating surface, and from Fig. 81 the evaporation will be about 7 or $7 \times 1.87 = 13.09$ pounds

of water per square foot of heating surface per hour, instead of 15 pounds as before. This would supply the cylinders at full stroke for 15 : 13.09 :: 10 : 8.7 miles an hour. From this, as a starting point, we construct a new hyperbola and tangent as before, which locus gives the combinations of speed and tractive force for 6,000 pounds of coal an hour. In the figure, the loci have been drawn for each thousand pounds per hour from 1,000 to 8,000.

14. If, now, we divide the quantities per hour by the speed, we obtain the fuel consumption per mile, and this is shown by

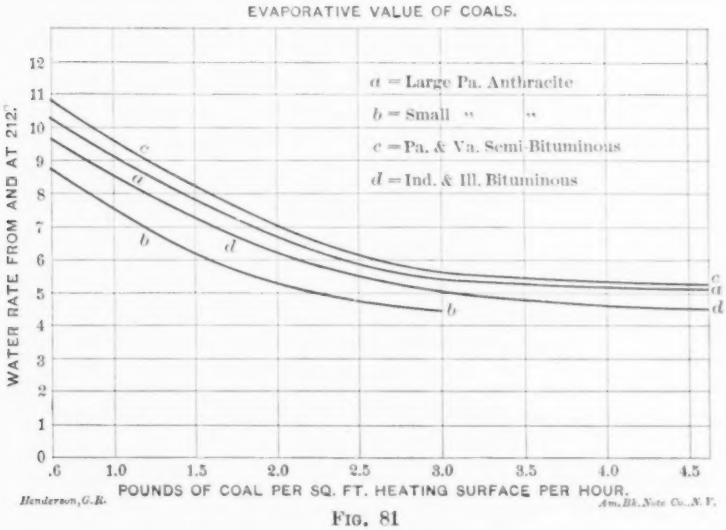


FIG. 81

the dotted lines. Thus with a tractive force of 20,000 pounds and a speed of 15 miles an hour, we should expect an hourly consumption of 3,000 pounds or 200 pounds per mile.

But as our diagram (Fig. 80) is based on speed and tractive force as co-ordinates, we can construct additional curves, which may represent the force needed to pull various trains at different speeds or up grades. For instance, if we consider a train of 1,000 tons weight back of tender, or 1,150 tons total weight, we can lay down the curves of resistance for a level, $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$ and 1 per cent. grade, as shown in the diagram. By this means we can determine the quantity of coal that would be used on different portions of a division.

If there were 40 miles each of level, ½ and 1 per cent. grade, the fuel consumption would run as follows:

40	miles	@	38	miles	an	hour,	210	lb.	per mile	= 8,400 lb.
40	"	@	24	"	"	"	330	"	"	= 13,200 "
40	"	@	17	"	"	"	490	"	"	= 19,600 "
				Total						41,200 lb.

It will be noticed that this considers the train to be run at the maximum speed which the engine can make on each grade or portion of track. If the speed be limited to 15 miles an hour uniformly, we should expect consumption as follows:

40	miles	on	level	@	50	pounds	per	mile	=	2,000 pounds.
40	"	"	½%	grade	@	190	pounds	per	mile	= 7,600 "
40	"	"	1%	"	@	430	"	"	"	= 17,200 "
				Total						26,800 "

15. This trip of 120 miles will, however, require 8 hours for its completion, whereas the first schedule is a trifle over 5 hours. The coal consumption in the first case is over 50 per cent. greater than in the second case. The effect of speed upon the coal pile is clearly shown by following any of the grade lines in the diagram. The rate per mile on a ½ per cent. grade will be about as given below:

Miles per hour.....	5	10	15	20
Coal per mile, lb.	150	160	185	250

16. The influence of loading is shown by the curves marked "Total tons up 1 per cent. grade."

At	10	miles	an	hour,	250	lb.	per	mile	for	1,000 tons
"	"	"	"	"	325	"	"	"	"	1,200 "
"	"	"	"	"	500	"	"	"	"	1,400 "

When we near the limit of capacity of the engine, the difference is still more marked.

At	10	miles	an	hour,	630	lb.	per	mile	for	1,500 tons.
"	"	"	"	"	680	"	"	"	"	1,520 "
"	"	"	"	"	720	"	"	"	"	1,540 "
"	"	"	"	"	780	"	"	"	"	1,560 "

Again, 1,600 tons can be hauled at 5 miles an hour with a consumption of 500 pounds per mile, whereas 1,570 tons at 10 miles

an hour will require 800 pounds per mile, both cases being the maximum that the engine can do.

17. These examples give an idea of the variety of problems in fuel consumption which can be quickly solved by the aid of this diagram. It is true that a different diagram must be made for each class of engine, but this is a comparatively small matter. With the increase in speed and loads of the present day, the coal consumption becomes a topic of great interest, and when comparisons are made with previous schedules, there is an apparent decrease in the economy of fuel, unless the various points are duly considered, and as explained by the chart.

DISCUSSION.

Prof. R. C. Carpenter.—The paper presented by Mr. G. R. Henderson gives a valuable method for rationally computing the fuel consumption of locomotives for different conditions of service and will, I believe, be found of permanent value for estimating the performance of locomotives. An interesting statement is made in paragraph 7 of the paper with reference to the limits of consumption of coal per square foot of grate area per hour. The limits are stated as follows:

Bituminous coal	200 lb.
Anthracite, large sizes	100 "
Anthracite, small sizes	60 "

These figures show the enormous quantities of coal which may be burned per hour in locomotive practice. These amounts are so large that they are not even approximated in any other case where coal is used, so far as I know.

During the past year I have had an opportunity to note the limits of combustion of several kinds of coal in stationary practice where a forced draft was employed, which was, however, very much less than the induced draft available in locomotive practice. In this particular instance the draft under the grate was maintained by a fan blower at from $1\frac{1}{2}$ to 2 inches of water pressure, the draft above the grate (suction) was approximately $\frac{1}{4}$ inch of water pressure. The bituminous coals used were slack obtained from northwestern Pennsylvania; the anthracites and semi-anthracites were obtained from the Lehigh Valley districts.

Under the conditions mentioned our experience indicates that the practical limits of coal consumption are about as follows:

Bituminous slack	28	lb. per sq. ft. of grate per hr.
Anthracite, Buckwheat No. 1	18	" " " " " " "
Anthracite, Buckwheat No. 2	14	" " " " " " "
Semi-anthracite, pea (Loyal Sock)	20	" " " " " " "
Semi-anthracite, dust (Loyal Sock)	12	" " " " " " "

The anthracite buckwheats, as above, contained in every case a considerable amount of dust, investigation usually showing not less than 25 per cent. and not over 40 per cent. that would pass through a screen with a mesh of $\frac{1}{16}$ inch, and would be classed as culm. Experience in burning small anthracite coals has shown a large variation in sizes shipped from the different mines under the same name, and an investigation has shown that no standard screen is employed in the different collieries, and consequently that there is great uncertainty as to the kind of coal which will be provided under any given name. A small variation in the size of the meshes in the screen employed, and also the shape of these meshes, whether round or square, makes a very great difference in the character of the coal supplied; therefore it would seem of great importance that the size and shape of meshes of the screen for each commercial size of coal should be definitely fixed by statutory provision. The late Eckley B. Coxe proposed the following table as showing the size of mesh through and over which pea coal and Nos. 1, 2 and 3 buckwheats were made in the Coxe Bros. Collieries:

Size of coal.	Over round hole.	Through a round hole.
Pea coal	$\frac{9}{16}$ in. dia.	$\frac{7}{8}$ in. dia.
No. 1 Buckwheat	$\frac{7}{8}$ " "	$1\frac{1}{16}$ " "
No. 2 "	$\frac{5}{16}$ " "	$\frac{7}{8}$ " "
No. 3 "	$\frac{3}{32}$ " "	$\frac{3}{16}$ " "

A recent investigation indicates that the Lehigh Valley collieries at the present time screen most of the No. 2 buckwheat by passing through a screen with a square hole or meshes $\frac{5}{16}$-inch, and over a screen with a square mesh or hole $\frac{3}{16}$-inch (on a side), and as a consequence the coals which are sold as No. 2 buckwheat by that company may be essentially different in size from those described in the table by Mr. Coxe.

The experience to which I referred, compared with the state-

ment in the paper, indicates that the relative amounts of the different kinds of coal which may be burned may be materially changed by changing the draft conditions; thus the relative amounts as stated by Mr. Henderson and as found by our tests are essentially as follows:

	Relative amount by Henderson.	Relative amount by our experience.
Bituminous coal	3.33	2.00
Anthracite, large sizes	1.66	....
Anthracite, small sizes	1.00	1.00

Prof. E. A. Hitchcock.—As I understand Mr. Henderson's treatment of this subject, it is largely theoretical in view of the fact that in computing the number of revolutions per minute that it is possible for the locomotive in question to make with lever in corner and boiler working to maximum capacity, he has taken the ultimate steaming capacity of the boiler and the theoretical quantity of steam used per stroke of piston, no allowance being made for cylinder condensation, which of course would be a large per cent., especially at low speed. Take the output of the boiler as given, 40,000 pounds steam per hour. The indicated horse-power with lever in corner and 180 pounds steam in cylinder and 60 revolutions per minute would be aproximately 1,200, thus giving 33.3 pounds steam per indicated horse-power per hour, a very low value for any engine working under such conditions. What would be the proper percentage correction to introduce for the case cited, and would not such correction change the location of the curves in Fig. 80?

Prof. W. F. M. Goss.—Any attempt to generalize concerning the fuel consumption of a locomotive, or its output of power, is likely to be regarded as of more than ordinary interest. In this respect, the paper of Mr. Henderson is no exception. The analysis he employs is interesting and ingenious. While it admits of but few refinements, it is probable that the conclusions reached, as set forth in Fig. 80, are sufficiently accurate for many practical purposes. When the results of the Pennsylvania Company's extensive work in locomotive testing are available, it should be possible to check or to correct them. But whether absolutely correct or not, the diagram is instructive in the relations it discloses. It shows at a glance very much that is fundamental and of general application concerning locomotive performance. For example,

the inclination of the principal curves indicates the rate at which draw-bar pull changes with changes of speed, and difference in the area between them represents the changes in efficiency resulting from variations in rate of power developed.

Mr. Harrington Emerson.—Engineering data are like an old map of Africa, much detail in certain quarters and black spaces in others. Much is known about steam and very little about hot gases; much about the heat units absorbed in water evaporation and almost nothing about the heat units absorbed in oil vaporization; much about the rapidity of heat flow from coal to water through boiler plate, very little about the rapidity of heat flow from internal combustion engine to water jacket; much about the combustion of coal, very little about the combustion of explosives under very high pressures; much about the specific gravity of solids at the earth's surface, nothing about the specific gravity of gases at the earth's center, where air and other gases are probably heavier than gold, this accounting for the high specific gravity of the earth. Nowhere in engineering are there greater blanks in definite knowledge than in railroading. The reason is not far to seek. Railroad men are overburdened with the daily exhausting and absorbing tasks and even when they reflect and speculate, the available appliances or methods to give accurate knowledge are lacking, and rarely have I had more occasion for wonder than in considering the great disparities in railroad practice. The length of track and grades are known to a tenth of an inch, but coal consumption is an unknown quantity. Trains are run to the minute, but a shop operation may dawdle through a year; bridges are repaired in a day, but I will take two years to dig a well or repair a leaking roof.

I therefore wish to thank the writer for his stimulating and interesting excursion into the dark region of locomotive fuel consumption, with varying loads and grades.

That one attempts to confirm or dispute his conclusions show to what extent the field is a new one and the work original. One matter he has left out. I do not find any curves indicating increased fuel consumption due to bucking snow drifts and keeping a long passenger train warm in 40 degrees below zero weather.

Mr. C. B. Rearick.—Referring to Professor Carpenter's remarks about the amount of coal burned per square foot of grate, I wish to state that five or six years ago I ran some tests on No. 1 Buckwheat coal fired under a vertical type boiler, and we burned

38 pounds of coal per square foot of grate with $\frac{3}{4}$ to $\frac{7}{8}$ inch natural draught, which was very remarkable, although it was repeated on two or three different tests; in fact, at one test 42½ pounds per square foot of grate was burned. The capacity of the boiler was about 1,000 horse-power.

It was a special type of boiler, which did not prove to be a very satisfactory steam unit for continuous service.

The grate used was made out of wire mesh and shaking type, and the boiler was fired by hand. The furnace was a rectangular and of brick. The boiler proper, the tube sheet, was six feet above the grates, and we carried a small layer of ash on the mesh and a light fire, watching it very closely.

Mr. A. A. Cary.—Referring to Professor Carpenter's statement regarding the amounts of coal which may be burned per square foot of grate, I have recently had my attention directed to a number of tests conducted at one of the mine plants of the Delaware, Lackawanna & Western Railroad, in which rice, barley and No. 1 Buckwheat anthracite coal were used.

In these tests 30 and more pounds of coal were burned per square foot of grate per hour by a special system using superheated steam under the grates.

The accuracy of these tests have been confirmed by the engineer of the contractor who installed the system and also by one of the engineers of the Delaware, Lackawanna & Western Co., but the former stated that the best results were obtained by their system when 25 pounds of such coal was burned per square foot of grate per hour.

The effect of superheated steam, used under the grates, with anthracite coal is a very interesting study. It certainly seems to keep the fire bed freer from large clinker than when saturated or moist steam is used, although one would think that the reverse would be the result.

I have occasionally heard surprise expressed concerning the greater amount of coal that can be burned per square foot of grate per hour when the more volatile coals of the West are used than when we use semi-bituminous or anthracite coal.

The reason for this becomes quite apparent when we recall the fact that all the combustible portion of anthracite coal consists practically of fixed carbon, which is burned directly upon the grates, while with bituminous coal which may, for this illustration, be considered as consisting of fixed carbon and volatile mat-

ter, the former burning directly upon the grate bars (the same as the anthracite), while the volatile matter burns as a gas above or beyond the fire bed.

Thus, supposing we are able to burn anthracite coal at the rate of 15 pounds of fixed carbon per square foot of grate, and then, with the same draft and furnace conditions, we use bituminous coal consisting of 50 per cent. fixed carbon and 50 per cent. volatile matter. We should then (roughly speaking) be able to burn twice as much bituminous coal as we formerly burned when using anthracite as no more fixed carbon would be burned per square foot of grate per hour with this amount of bituminous coal.

Of course, we must provide larger combustion chambers when using coals carrying volatile matter than when using anthracite, to provide for a proper combustion of the gaseous matter, and the combustion chamber must be properly designed to maintain the ignition temperature of the gases until they are entirely consumed.

I have not, thus far, taken into account the ash, which is a very important factor in the furnace, clogging the air passages through the fuel bed more and more as its percentage increases per pound of fixed carbon contained in the coal.

Mr. O. C. Woolson.—I am a little surprised to learn that Mr. Cary has been able to obtain these tests of the Delaware and Lackawanna road, because the impression I have is that those tests are made for the consumers of coal, and as soon as it becomes apparent to the Delaware road that an expert would like to look them over they shrink up—they go in their holes right away; and I have heard it remarked that those tests were made particularly for the people who are a little inclined to use soft coal for good and practical reasons, and to induce them to use hard coal they have established certain proportions of hard and soft coal mixtures to get enormous results; and therefore, if they were aware that Mr. Cary was an expert in the matter, I don't see how he got it.

Mr. A. A. Cary.—The tests referred to by Mr. Woolson are not the same as those I have spoken of.

In the tests I referred to, small-sized anthracite coals alone were used, and I have every reason to believe that the rate of combustion I have mentioned was obtained.

Mr. Wm. O. Webber.—In paragraph 18 on the last page of this paper it is very interesting to note the losses due to unburnt

coal. One case being nearly 15 per cent. and the other slightly over 10½ per cent. I would like to ask if there is any data as to the loss on western roads using bituminous coal?

Mr. G. I. Rockwood.—I would like to ask Mr. Henderson how much coal a fireman can be expected to handle on a trip? We will say we have a long train and a long trip. How many tons of coal per hour, and for how many hours can he keep it up? We hear of so many complaints from firemen in stationary plants. I should like to know what his experience is.

Mr. Webber.—I used to shovel about five tons of coal in four hours.

Mr. Wm. Kent.—Regarding Mr. Webber's question, I think that is very fully gone into.

*Mr. G. R. Henderson.**—There are several points brought out in the notes of Professor Carpenter. I think he strikes the keynote regarding the rate of combustion, as it is enormous and spasmodic in a locomotive.

In regard to the cylinder condensation spoken of by Mr. Hitchcock, I wish to say that these curves have been checked up by actual tests, and agree fairly well, so I think that has pretty well been covered. The tests gave about 33 pounds steam per indicated horse-power under the conditions mentioned.

The combustion spoken of by Mr. Cary of 30 pounds of buckwheat coal with a steam jet is below the maximum on a locomotive, where we have a steam jet of great power in the exhaust up the stack.

Professor Goss' book "Sparks," issued about two or three years ago, gives very full information regarding the proportion of cinders and unburnt fuel emitted from the stack of a locomotive.

In regard to the question of how much coal a fireman can handle, that is what is agitating the railroads nowadays. In tests on the Chicago and North Western we made a record of shovelling about six thousand pounds per hour, and that was about all one fireman could do. The engine was not on the road, but in the test plant. The coal was wheeled to him and he was kept busy putting the coal in.

I have stated in the paper that it was a question whether 8,000 pounds of coal per hour could be handled by one man, but then there is a mechanical stoker for handling coal with double that

* Author's Closure, under the Rules.

capacity. However, that is another question. This paper is on combustion and not on the capacity of firemen. I think it is pretty generally admitted now that three tons an hour is about the limit that a man can handle and properly distribute in the firebox of a locomotive.

No. 1067.*

A NEW HYDRAULIC EXPERIMENT.

BY A. F. NAGLE, BUFFALO, N. Y.
(Member of the Society.)

1. The determination is that of the rise of water in a standpipe having a long supply pipe, when the draft suddenly ceases.

2. In the case of the standpipe supplied by a long pipe, or tunnel, from a distant reservoir, and from which standpipe pumping engines are supplied, the question of how high the water will rise in the said standpipe, if the engines should suddenly cease to operate, is one on which there has been much discussion, but, so far as I am aware, no experiments have ever been made which would tend to confirm or contradict any theory applied to the problem.

3. Not with the view of establishing a full and correct formula for the solution of the above problem, but simply to set at rest a preconceived notion that under no conditions could the water rise above its original level in the reservoir, I made several experiments which may be of interest to our membership. I trust, however, that these simple experiments may lead others to institute more exhaustive tests for the purpose of determining a scientific formula, based upon experimental data, for the solution of this problem.

4. The apparatus is shown in Fig. 82. It consists of an ordinary oil barrel, kept full to overflowing with water from a hose, the surface of the water in the barrel being kept 46 inches above the floor. First, a $\frac{1}{2}$ inch wrought-iron pipe, 48 feet 2 inches long, containing three couplings, was run to within 10 inches of the floor, followed by an ordinary cast iron elbow and vertical riser. In this riser and 5 inches above the pipe, a cast-iron tee was inserted, on the side of which tee a common bib cock was attached

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*

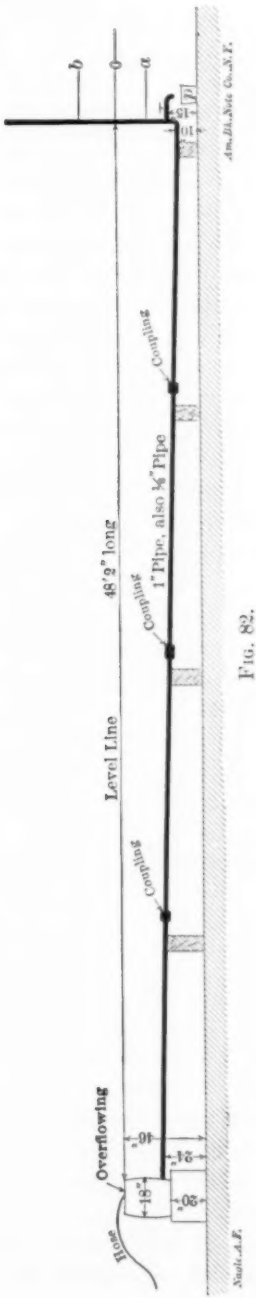

b
0
a
Coupling
1" Pipe, also ¾" Pipe
48'9" long
Coupling
Level Line
Coupling
Overflowing
Hose

Fig. 82.

for the purpose of drawing water into the pail. Above the tee was attached an ordinary glass water tube, 0.60 inch internal diameter.

5. The first set of experiments were made as follows: When the water had come to rest, an elastic was placed on the glass tube to mark the position of the level line. Another elastic was set 24 inches below the level line, and the bib cock was opened and adjusted until the water was drawn down to it and a constant flow established. The cock was fixed, with the water in the tube at this lower position, and the water escaping under these conditions was then taken up in pails and weighed each minute. These experiments were repeated, but they ran very uniformly.

6. Then the bib-cock was closed suddenly and the rise of water in the glass above the level line was noted. These results varied somewhat, and three or more experiments were made in each case, but the results given are fair averages.

7. Experiments of a like nature were made with the water under constant flows and at levels in the glass tube of 18 inches, 12 inches and 6 respectively below the level line. These results are given in Table 1 and plotted in Fig. 83. The calculations for velocity of flow, column 4, are made from the weight of water discharged and the known diameter of the pipe 0.625 inch. Column 5 is obtained from the well-known formula

$$V^2 = 2\,gh \text{ or } h = \frac{V^2}{64.32} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Column 6 is column 5 reduced to inches.

Column 7 is the total constructed head in inches below the level line, and is, of course, the total loss of head H, as found by the formula

$$H = (8 + \frac{L}{d}\,z) \times h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

H = total head.
h = velocity head.
L = length of pipe in feet.
d = diameter of pipe in feet.
z = coefficient of friction.

Column 8 is the observed rise above the level line in inches.

Column 9 is calculated from formula (2) and solved for z, the coefficient of friction.

8. For the resistance of inflow, couplings, elbow and tee, I have assumed the value of eight (8) times the velocity head h in the

case of the ½-inch pipe, and five (5) times in the case of the 1-inch pipe. I have made this calculation as a check upon the general results found, and in a general way it is confirmatory of the reliability of the experiments.

Column 10 is a theoretical value for column 8.

9. The second experiment consisted in substituting an inch (1-inch) pipe for the one-half (½-inch) up to the point of the bib-cock. The cock was so small, and the glass tube so much smaller than the size of the pipe, that I abandoned that experiment for all purposes of calculation, although the data obtained are given in Table II and plotted in Fig. 84.

10. The third experiment consisted in substituting an inch wrought-iron vertical pipe for the small glass tube, and also putting in a full inch plug-cock for the smaller ½-inch bib-cock. I was unable to find a glass tube of the size required, hence I used the iron pipe, and, after a little experience, found that a large cork float with a thin long stick of wood gave prompt and satisfactory indications of the water level in the riser tube. The methods of procedure were precisely the same as had been followed with the ½-inch pipe, and the calculations made were the same, except that instead of allowing eight (8) times the velocity head for various losses, only five (5) times the velocity head was used. The results are given in Table III and plotted in Fig. 85.

11. A careful study of the values of the coefficients of friction, z, does not indicate any material difference due to different velocities. Grouping them together we have

Velocity	1.84	ft.	per	sec.	1	in.	pipe,	$z = .04250$
"	1.70	"	"	"	½	"	"	$= .03913$
"	1.60	"	"	"	1	"	"	$= .03620$
"	1.48	"	"	"	½	"	"	$= .03863$
"	1.15	"	"	"	½	"	"	$= .04353$
"	1.13	"	"	"	1	"	"	$= .03620$
"	.85	"	"	"	½	"	"	$= .03910$
Averages	1.39	"	"	"				.03933

12. Is it possible to deduce a formula from the above data applicable to the general problem? The theoretical formula, ignoring friction or other losses, and when the standpipe is of the same diameter as the main supply pipe or tunnel, is very simple. In that case the residual energy of the water in the long pipe must be absorbed by the work done in lifting a column of water

Vertical Glass Tube. .60" diameter Inside.
Vertical Pipe 1" (W.I.) with cork float and thin stick of wood.
Level line
Theoretical Height in Tube (Neglecting all Friction)
Actual Height in Tube
Line B.
Velocity in 1" Pipe in ft. per sec.
Velocity in ¾" Pipe in ft. per sec.
Fig. 83.
Fig. 84.

in the standpipe to a mean height of one-half its own height. This would be expressed by the formula

$$L \times h = X \times \frac{X}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where L = length of pipe or tunnel,

h = head due to velocity $= \frac{V^2}{2g}$,

X = rise in standpipe above level of reservoir.

Equation (3) can be expressed as

$$X = \sqrt{2\,Lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or substituting the valve of h in terms of v, we have

$$X = 0.176\, v \sqrt{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Applying the formula to the experiments in Tables I and III taking $L = 49$ feet, $\sqrt{L} = 7$ feet, we find the values for X, reduced to inches, in Column 10.

TABLE I.

HALF-INCH WROUGHT IRON PIPE WITH GLASS TUBE STANDPIPE.

$\frac{L}{d} = 933.$

1	2	3	4	5	6	7	8	9	10
Number of Experiment.	Pounds of Water per Minute.	Cubic Inches of Water per Second.	Velocity in Feet per Second.	Velocity Head in Feet.	Velocity Head in Inches.	Water Below Level Line in Inches.	Water Above Level Line in Inches.	Coefficient of Friction.	Theoretical Height Above Level Line.
									Inches.
1..	13.25	6.12	1.70	.0449	.539	24	12.00	.03913	25.13
2..	11.75	5.42	1.48	.0341	.409	18	11.25	.03863	21.87
3..	9.08	4.19	1.15	.0206	.247	12	10.50	.04353	17.00
4..	6.75	3.11	.85	.0112	.135	6	9.00	.03910	12.56
								Mean .0401	

TABLE II.

ONE-INCH WROUGHT IRON PIPE WITH GLASS TUBE STANDPIPE.

1	2	3	4	5	6	7	8	9	
5..	18.25	8.42	.81	.....	.....	12	30	.....	
6..	16.37	7.56	.73	.....	.....	9	27	.....	
7..	12.87	5.94	.57	.....	.....	6	33	.....	

TABLE III.

ONE-INCH WROUGHT IRON PIPE WITH WROUGHT IRON STANDPIPE.

$\frac{L}{d} = 555.$

1	2	3	4	5	6	7	8	9	10
8..	41.25	19.04	1.84	.0526	.631	18	19	.0425	27.20
9..	36.00	16.62	1.60	.0398	.477	12	16	.0362	23.65
10.	25.50	11.74	1.13	.0198	.237	6	12	.0362	16.70
								Mean .0382	

13. These values of X are plotted in Figs. 83 and 85, and reveal the difficulty of constructing a formula which will give results fitting the experiments noted.

It is, perhaps, reasonable that in the case of the smaller pipe the greater deviation from the theoretical rise should be found, but the indications are that in the larger sizes the rise will approach close to the theoretical height.

14. To illustrate the application of the formula. Assume a 7-foot diameter tunnel, 24,000 feet long, supplying 100 million gallons of water per 24 hours, causing a velocity of 4 feet per second.

If the standpipe is of the same diameter as the tunnel, the theoretical height to which the water may rise will be $0.176 \times 4 \times \sqrt{24{,}000} = 0.704 \times 155 = 109$ feet.

Should the standpipe be enlarged to 12 feet diameter, the equivalent height to which the water will rise can be found by the formula.

$$d^2 \times h \times \frac{h}{2} = D^2 \times H \times \frac{H}{2} \text{ or } d \times h = D \times H$$

where d = original diameter of pipe,
h = height of column as found by equation (5),
D = enlarged diameter,
H = height in enlarged pipe.

Making these substitutions we have

$$0.7 \times 109 = 12 \times H \text{ or}$$
$$H = 64 \text{ ft.}$$

The case of a 24-inch pipe, 26,000 feet long, velocity of water 3.50 feet per second, and having a 10-foot standpipe, is given me.

15. The theoretical rise in the standpipe if it were also 24 inches in diameter is, according to formula (5) equal to

$$0.176 \times 3.50 \times \sqrt{26{,}000} \text{ or } 0.616 \times 161 = 100 \text{ feet.}$$

To reduce this to a 10-foot diameter standpipe we have—

$$d^2 \times \frac{h^2}{2} = D^2 \times \frac{H^2}{2} \text{ or } d \times h = D \times H,$$

$$2 \times 100 = 10 \times H.$$

$$H = 20 \text{ feet.}$$

DISCUSSION.

Mr. George W. Colles.—Attention should be called to the fact that the height to which the water will be raised in the standpipe by the sudden shutting off of flow represents the maximum pressure at the base of the standpipe only in cases in which there is little or no head of water in the standpipe when the water is flowing. It is true the author says nothing to the contrary; but of what use are the conclusions of the paper if not to exhibit the maximum possible pressure in the pipe?—to determine which, however, the author's equations are in most practical cases useless, or at most but partial, as the problem must be approached from a different direction.

The maximum pressure in the pipe caused by the shutting off of the water will of course occur at the end of the moving body of water—at the junction of the moving and stationary columns, if a standpipe be present. Designating by L and V the length and velocity of the moving column respectively, by l the length of the stationary column at the epoch of steady flow, by v the common velocity just after the flow has been shut off, and by t the time from the commencement of the shutting off till the two columns are assimilated to the velocity v, we have, as the joint momentum of columns of unit mass per unit length,

$$LV = (L + l)\, v,$$

whence

$$v = \frac{LV}{L + l}$$

The force f necessary to produce this change of velocity in each of the columns will naturally depend on the time t in which it

takes place. According to the well known equation of forces, we have

$$f = \frac{lv}{t} = \frac{l}{t} \cdot \frac{LV}{L + l}$$

and this force must be added to the head of water l existing in the standpipe during flow in order to obtain the total pressure (neglecting friction) to which the pipe is subjected. The quantity f/g is in terms of head, so that we have for the total head

$$H = \frac{f}{g} + l = \frac{l}{gt} \cdot \frac{LV}{L + l} + l.$$

In the cases adduced in the paper, the head, l in the standpipe is practically nothing, so that the momentary head, H, is also trifling; but suppose, in Experiment 8 (Table III.), for instance, a head of 50 feet were present in the standpipe during flow, as would be nothing unusual in practice; then $l = 50$, $L = 48$, $V = 1.84$, and

$$H = \frac{l}{t} \times \frac{50 \times 48 \times 1.84}{32.2 \times 98} + 50 = \frac{1.45}{t} + 50.$$

If it requires $\frac{1}{10}$ second to shut off the water, then $H = 64.5$ feet, that is, an excess of $14\frac{1}{2}$ feet, as against 3 feet 9 inches theoretically required by the author's equation, which is independent of the head in the standpipe.

To the case of an ordinary water-pipe, which has no standpipe attached, the equations and examples of the paper have no application whatever, and are therefore not merely useless but misleading. In this case the head generated by the sudden shutting off of the water becomes

$$H = \frac{LV}{gt},$$

there being no release for the imprisoned water, which must be brought to a dead stop in a very limited space of time. Taking the same case as before, but removing or plugging the standpipe, we should have, therefore, $L = 48$, $V = 1.84$, $t = 0.1$, and

$$H = \frac{48 \times 1.84}{32.2 \times 0.1} = 27.5 \text{ feet},$$

which of course must be added to the head previously existing at the same point.

It will be seen, therefore, that the cases occurring in practice cannot be figured from the equations given in the paper, as the

pressures attained depend on wholly different considerations. For example, a long turbine penstock, in which the water is allowed to reach considerable velocity, and is then suddenly shut off or violently checked at the lower end, would burst if not provided with relief valves, although the height to which the water

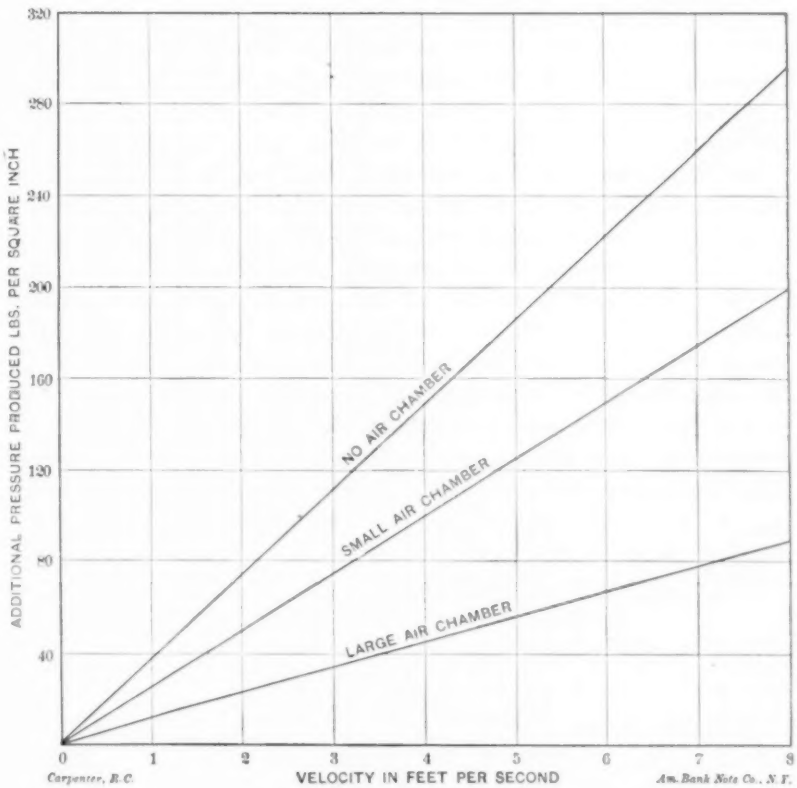


FIG. 86.

would rise in a standpipe of enlarged diameter would be insignificant.

Prof. R. C. Carpenter.—The experiment which Mr. Nagle describes is essentially of the same general nature as one described by the writer in Vol. XV, A.S.M.E., under the title of "Some Experiments on the Effect of Water Hammer." I introduce a diagram showing the general results of that test (Fig. 86). The experiments were made on a two-inch pipe through which the

water was allowed to flow at various velocities which were measured with a Pitot tube (Fig. 87). A valve was connected to the pipe which could be suddenly opened and closed as desired. The static pressure on the pipe averaged very closely to 27 pounds. The resultant pressure produced by the sudden closing of the valve depended upon the velocity of the water flowing, how suddenly stopped, and also as to whether or not an air chamber was supplied. This pressure, in case of the pipe with no air chamber, in some cases went to 300 pounds, or more than ten times the initial static pressure. In the test referred to the pressures were measured by use of a delicate indicator.

In the discussion of that paper the late Prof. DeVolson Wood presented a formula for theoretical consideration which was not, however, entirely satisfactory. Quite recently this matter has been very thoroughly investigated by Prof. M. Joukowsky at Moscow, and the results of all the investigations have been compiled by Mr. O. Simin in a paper which is soon to be published by the Association of Engineering Societies. The Joukowsky experiments were made on large pipes from 1,000 to2,500 feet in length and were very exhaustive. Joukowsky deduces a formula for the velocity of propagation of the pressure wave and also for the maximum additional pressure produced by the closure of the gate. The formulæ produced are rational and agree very closely with the experimental results. The formula for the velocity of wave propagation is

$$(1) \quad \ldots \quad \lambda = \frac{1}{\sqrt{\frac{\gamma}{K g} + \frac{2 R_\circ \gamma}{e E g}}} = \frac{1}{\sqrt{\frac{\rho_\circ}{K} + \frac{2 R_\circ \rho_\circ}{e E}}},$$

where $2 R_\circ$ = diameter of the pipe.

e = thickness of the walls of the pipe.

E = Young's linear modulus of elasticity for the material of the pipe = about 1,000,000 kilo. per square centimeter for cast iron.

K = volumnar modulus of elasticity of water = about 300,000 pounds per square inch.

γ = weight of unit of volume of water.

g = acceleration gravity.

$\rho_\circ = \frac{\gamma}{g}$ = density of water.

The formula for the maximum additional pressure, P, produced by the closure of the gate, is as follows:

$$(2) \quad \ldots\ldots\ldots P = \frac{v \lambda \gamma}{g} = v \lambda \rho_{\circ}.$$

where v = extinguished velocity of water in the pipe.

λ = speed of propagation of the pressure wave in the pipe.

The other symbols as in the preceding formula.

From these formulæ we notice that the velocity of wave propagation depends on the size, thickness and kind of pipe, and on the density and modulus of elasticity of the liquid. The increase, P, of pressure varies in direct proportion to the extinguished velocity of flow, is directly proportional to the velocity of propagation of the pressure wave and of the density of the liquid. it is independent of the length of the pipe.

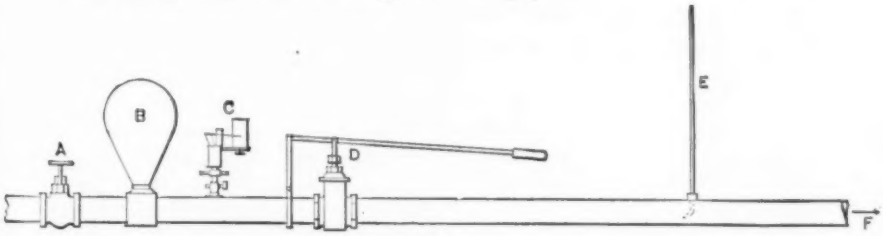


FIG. 87.

The following table shows the calculated and observed velocity of the wave propagation in feet per second in cast-iron pipe.

Diameter of pipe, inches.	Velocity in feet per second, calculated.	Velocity in feet per second, observed.
2	4,424	4,375
4	4,228	4,200
6	4,116	4,100
24	2,996	3,313

The following table gives the calculated value of additional pressure produced by suddenly arresting the flow in pipes of different diameters in terms of velocity of flow in feet per second.

Diameter of pipe, inches.	Additional pressure for each foot per second of extinguished velocity of flow in atmospheres.
2	4.066
4	3.886
6	3.783
24	2.754

From the table it is seen that increased pressure in small pipes (2 to 6 inches in diameter), for each foot of extinguished velocity of flow, equals about 4 atmospheres or 60 pounds.

The experimental results measured with a very delicate and accurate apparatus agree closely with the calculated results, and substantially prove that the formulæ deduced are practically accurate. This theory and also the experiments indicate that under certain conditions when the velocity is arrested in a dead ended pipe the wave motion from one end may be superimposed with that from the other and a very high pressure may result.

Experiments also indicate that the intensity of the shock or additional pressure produced will be lessened by slowly arresting the velocity of flow, and can probably be entirely obviated by closing the gate very slowly. The relation which the time of closing of the gate bears to the pressures produced is expressed by Joukowsky by the following formula:

$$t = \frac{P}{P^1} \cdot \frac{2\,l}{\lambda} = \frac{v\,h}{P^1} \cdot \frac{2\,l}{\lambda},$$

where

t = duration of closure of the gate in seconds.
v = velocity of flow of water.
h = pressure obtained per unit of extinguished velocity of water, v.
$P = v\,h$ = full maximum shock pressure.
P^1 = the permissable shock pressure.
l = length of pipe line.
λ = velocity of the propulsion of the pressure wave.

Sir Benjamin Baker.—Some 20 years ago I was consulted about repeated fractures in a large cast-iron pipe about 3 miles in length with pumps at the lower end driven sometimes by turbines and sometimes by steam engines, and delivering at the upper end into an open reservoir about 90 feet above the level of the pumps. The fractures occurred about 60 feet from the engines under a public road. Very large air vessels were introduced and the engines modified in a variety of ways without effect. He ascertained that the fractures never occurred when the pumps were worked by the turbines, so instead of advising a new type of pump, as the authorities expected, he merely ordered a screw

action starting valve for the engine similar to that on the turbines, so that the engines could not be started or stopped in less than a minute. No further fractures occurred, as the latter obviously were due to the column of water continuing to flow on after the pumps were suddenly stopped and then returning with a water-hammer action on the pipe at its lower end.

Mr. H. H. Suplee.—Referring to the experiments made by Joukowsky in Russia upon hydraulic shock or water-hammer in pipes, I think that especial attention should be given to them, as they exceed in magnitude and scale anything that has been done anywhere in this little-studied question. The original memoir of Professor N. Joukowsky, of the Imperial Technical School of Moscow, was published in the Transactions of the St. Petersburg Academy of Sciences in Russian in 1898, and again in German in 1900, the latter, a large pamphlet, being entitled "Ueber den Hydraulischen Stoss in Wasserleitungsröhren," and to be had at the price of 4 marks, from G. Haessel, Leipzig. A translation has recently been made by Miss Olga Simin, daughter of an eminent Russian hydraulic engineer, Mr. Nicholas Simin, and published in the Transactions of the American Waterworks Association for 1904.

Mr. A. C. Walworth.—It seems to me that this is practically a hydraulic ram. With such a ram we could, were it not for frictional and other losses, raise one-tenth of the water used to a height equal to ten times the difference of level between the water in the reservoir and the check valve when the water issues from the ram. We have always thought it a *very* wasteful machine, but the figures given us by the manufacturers are surprising; so that a ram may raise 1-17 of the water to ten times the height of the fall, an efficiency of about 50 per cent. Figures have been made by the makers which give the exact percentage.

*Mr. John C. Trautwine, Jr.**—Noting your paper, entitled "A New Hydraulic Experiment," which was presented at the Annual Meeting of the American Society of Mechanical Engineers in New York, December, 1904, I venture to bring out a certain point from a paper on Water Hammer, prepared by Miss Olga Simin, and based upon extensive experiments made at Moscow, Russia, in 1897.

The experiments were made on considerable lengths of pipe,

* Non-member.

sufficient to permit observations of the travel of the pressure wave.

It appears that if the pipe is so long, or the time of closing the gate is so short, that the pressure wave has not more than time to travel from the gate to the origin and back to the gate during the closing of the gate, then the full maximum pressure, P, due to the closure of the gate and $= \frac{v \lambda \gamma}{g}$, will be felt in some part or all of the pipe, no matter how slowly (within the limit mentioned) the gate be closed. But if the pipe is so short, or the time of closing the gate is so long that the pressure wave has *more* than time to travel from the gate to the origin and back to the gate, then the maximum pressure, $P = \frac{v \lambda \gamma}{g}$, will not be reached in any part of the pipe.

In this latter case, of course, the more slowly the gate be closed (for a given length of pipe), or the longer the pipe (for a given duration of closure), the less will be the maximum pressure reached.

The application of the principles established by these experiments would, I think, enable us to determine in advance the height to which the water would rise in your standpipe in any given case, the necessary data being given.

*Mr. A. F. Nagle.**—This problem, which I have attempted to solve, is an eminently practical one. The example given is an actual case where a standpipe 12 feet in diameter has been built to a height of 46 feet above the level of the water in a reservoir 23,000 feet distant and with which it is connected by a tunnel 7 feet in diameter. Provision will be made, however, for an overflow. A branch to the pumps is taken from the standpipe at a point 40 feet below the reservoir level. My theoretical formula gives the rise of water in this standpipe above the reservoir level as 64 feet, if the pumps should all suddenly cease to operate, a contingency not likely to arise, but one well to provide for. While it is true that the actual rise must be less, I confess I do not know how to introduce the factors for friction into this formula. The experiments with the 1-inch pipe show the actual rise to be 70 per cent of the theoretical, quite uniformly for the three experiments, but those obtained with the ½-inch pipe (viz., 48 per cent., 51 per cent., 62 per cent., and 71 per cent.), do not show any

* Author's Closure, under the Rules.

regularity in this ratio. This fact precludes the finding even of an empirical formula, and I am, therefore, driven to question the reliability of the ½-inch pipe experiments. I was hoping some member would bring forward a formula which would take into account the friction, but instead of doing this, several gentlemen have discussed water-hammer and resulting pressures, subjects with which I do not deal.

No. 1068.*

ROAD TESTS OF BROOKS PASSENGER LOCOMOTIVES.

BY E. A. HITCHCOCK, COLUMBUS, O.
(Member of the Society.)

1. This paper is presented to the Society because the writer believes it will be of some importance to those members interested in locomotive performance in that the results obtained represent actual every-day practice on two types of machines, and also because it supplements some trials on freight locomotives conducted on the same road and in nearly the same manner, the results of which were presented at the Chicago meeting of the Society, one special point of interest for all cases being the boiler-heat balance.

2. As the Hocking Valley Railway Company wished to draw comparisons between the two types of engines used at the time in their passenger service between Columbus and Toledo, it was decided to conduct at least four trials under as near as possible identical conditions, two on each locomotive.

3. These trials were carried on as thesis work by Messrs. E. R. Feicht, F. Singleton, R. B. Smith and B. S. Watters, members of the 1904 class in Mechanical Engineering of Ohio State University, ably assisted by R. E. Rightmire, representing the railroad company.

4. The locomotives were of Brooks make, dimensions of which are given in the following pages, the greatest difference, however, being the Belpaire type of boiler, with 72-inch drivers on No. 73, and the Wagontop boiler, with 66-inch drivers, on No. 80. No. 73 had been in service several years, but was thoroughly overhauled nine months previous to the trials, and had been in continuous service from that time, while No. 80 was practically new, having been in service only two months before her trial. In each case,

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

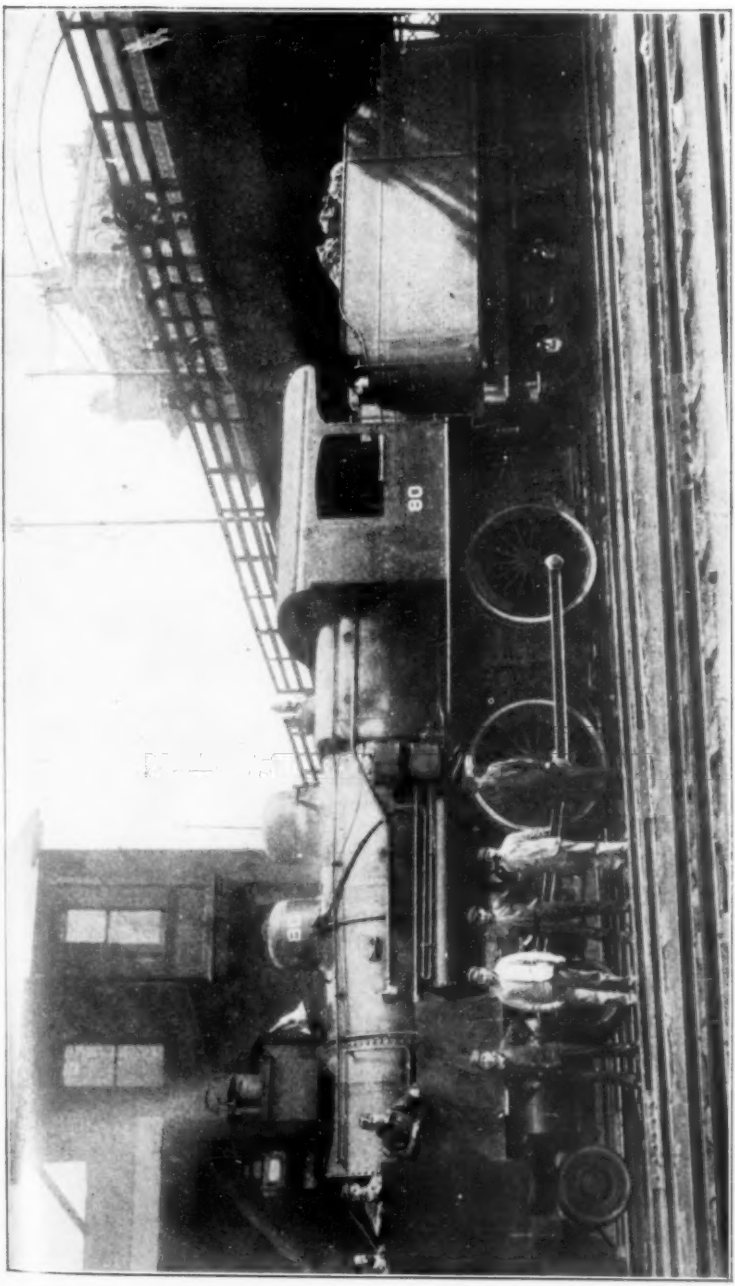


FIG. 88.

however, where needed, the engines were given new piston rings, the valve seats were put in good condition and valves reset so that each engine worked "square."

Apparatus and Preparation.

5. The apparatus used, although practically the same as on the trials of the consolidated freight engine the previous years, was as

FIG. 89.

follows: A gauge glass and scale on each side of each tank for determining the amount of feed water. A Barrus calorimeter connected to the dome. A Hohmann and Maurer mercury pyrometer inserted in front end. Draft gauges in the cab to indicate fire drafts at front end, fire-box and ash pan. The same flue gas-sampling and collecting apparatus was used as in previous year, and also the same combustible collector at the stack. A continuous counter was driven from the reducing motion, and also an indicating tachometer from the forward truck axle. Two indicators were used on each cylinder, with ¾-inch pipes, 9 inches long, well lagged, connecting to each end of each cylinder. The

indicators used were American Thomson and Star Brass outside spring. Slotted pendulum reducing motion was used, with exception of on one side of No. 80, where a pantograph was substituted on account of the air tank interfering with the other form of reducing motion, this tank being located underneath the running board. A continuous stroke counter connected with the air pump,

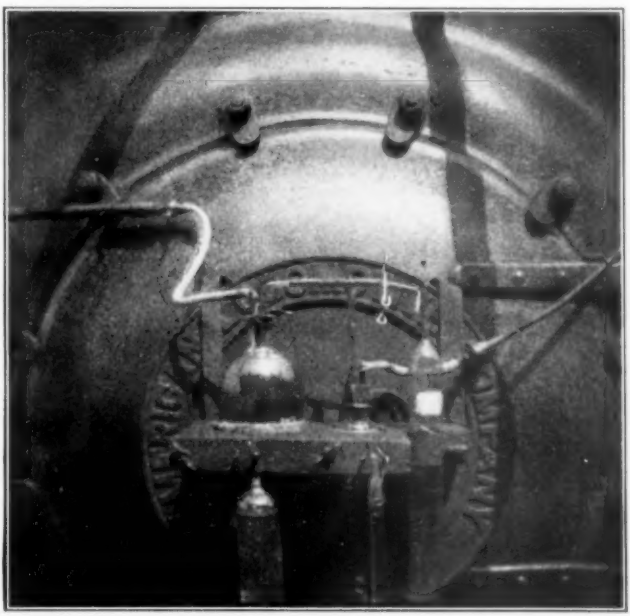


FIG. 90.

receiving its motion from a small plunger working in a brass cylinder.

6. Figs. 88, 89 and 90 show the form of wind shield used, its wings extending back past the cylinders, all serving a very good protection; the stack-refuse collector bolted to side of the stack and the flue-gas sampling and collecting apparatus fastened to the name plate on front of boiler.

Method of Conducting Trials.

7. Since these trials extended over the whole division, a distance of 124 miles, the total amount of coal used from the time the fires were first started in raising steam until the end of the run

was easily obtained. The thickness of the fire was noted at the commencement of the trial, and also the amount of coal used up to that time, when at end of trial the fire was left as near as possible in same condition. It was then allowed to burn out, then dumped and the ash pan thoroughly cleaned, so that the refuse obtained in the ash pan was from the total coal burned. The sack method was used for determining accurately the amount of coal used.

8. The number of injector applications were noted, so as to make correction for loss at overflow, also the number of times and length of popping of safety valve, the loss of steam in this direction being determined from a test of the pop valve while in place on the engine.

9. The steam used from the air pump was computed from the data furnished by the New York Air Brake Company.

10. All readings and indicator cards were taken at intervals of four minutes, with the exception of the sample of flue gas, which was taken practically every eight minutes.

11. On account of the risk involved, it was considered advisable not to attempt to read the calorimeter except when going into the station or in pulling out; so that, in all probabilities, there would be some variation between the amount of moisture determined and the true average.

12. The stack-refuse sampling apparatus was so constructed that the collecting pipe could be fastened at three different points on a radius of the stack. It was clamped in each of those positions for practically one-third of the run.

General Dimensions.

H. V. Passenger Locomotive.	No. 73.	No. 80.
Weight on drivers	84,000 lb.	87,000 lb.
" truck wheels	44,000 "	46,000 "
Total weight	128,000 "	133,000 "
Weight of tender	95,000 "	102,000 "
Wheel base, rigid	8 ft. 9 in.	8 ft. 9 in.
" " total of engine	24 " 1½ "	24 " 0 "
Length of engine	36 " 1½ "	36 " 0½ "
" " and tender	59 " 3½ "	59 " 4 "
Height of center of boiler above rails	8 " 7½ "	8 " 4½ "
" top of stack above rails	14 " 11 "	14 " 8 "
Heating surface, fire-box	161 sq. ft.	170 sq. ft.
" " tubes	1,813 "	1,776 "
" " total	1,974 "	1,946 "

	No. 73.	No. 80.
Grate area	30.8 sq. ft.	29 sq. ft.
Drivers, diameter in inches	72 in.	66 in.
" material of centers	cast-steel.	cast-steel.
Main wheel fit	8⅝ in.	8⅝ in.
Truck wheels, diameter	36 "	30 "
Journals, driving axle, diameter	8½ in. x 11 in.	8½ in. x 11 in.
" truck axle, "	5½ in. x 12 "	5½ in. x 12 "
Main crank-pin, diameter	5½ "	5½ "
" " length	5½ "	5½ "
Cylinder diameter	18 1/2 in.	18 "
Piston stroke	26 in.	26 "
" rod diameter	3½ "	3½ "
Kind of piston rod packing	Jerome.	Downing.
Length of main rod, center to center	7 ft. 9 in.	7 ft. 9 in.
Steam ports, length	17 in.	17 in.
" " width	1¾ "	1¾ "
Exhaust ports, length	17 "	17 "
" " width	3 "	3 "
Bridge, width	1¼ "	1¾ "
Valves, kind	American.	Richardson.
" greatest travel	6¼ in.	6½ in.
" outside lap	1 "	1 "
" inside lap	0 "	0 "
Boiler, type	Belpaire.	Wagon Top.
Working steam pressure	180 lb.	180 lb.
Material in barrel	carbon Steel.	Otis steel.
Thickness of material	7/16", 5/8", 7/16" & ½"	5/8", 5/8", 7/16" & ½"
Barrel, outside diameters	62 in.	62¼ in.
Kind of seams, longitudinal	quintuple lap.	sextuple butt.
" " circumferential	double lap.	double lap.
Tube sheet, thickness	¾ in.	⅝ in.
Kind of crown sheet stays	Tennessee iron.	Tennessee iron
Dome, diameter outside	31½ in.	31½ in.
Fire-box, length	108 in.	107½ in.
" width	42 "	39½ "
" depth, front	75 "	75 "
" " back	57 "	62 "
" material	carbon steel.	Otis steel.
" thickness of sheets	3/8", 7/16", 5/16", 5/16"	3/8", 7/16", 5/16", 5/16"
" width of water space	4 in. front, sides and back 3½ in.	4 in. all around
Grate, length	107 in.	106¾ in.
" width	34 "	34 "
" kind	Rocking finger.	Rocking finger.
" number of bars	10	10
Tubes, number	301	295
" material	Shelby steel.	Shelby steel.
" outside diameter	2 in.	2 in.
" length over sheets	11 ft. 7 1/16 in.	11 ft. 7 1/16 in.
Smoke-box, outside diameter	65 in.	65 in.

	No. 73.	No. 80.
Smoke-box, length	60 in.	60 in.
Exhaust nozzle, single or double	Single.	Single.
" " fixed or variable	fixed.	fixed.
" " diameter	4⅝ in.	5 in.
Distance of tip from center of boiler	2½ in., above.	7 in., below.
Netting, size of mesh	¼ in. x 1¼ in.	¼ in. x 1½ in.
Stack, kind	tank steel.	cast-iron, taper
" diameter	12½ in. & 15⅞"	14½ in & 17⅞ in.
" height above smoke-box	43 in.	43 in.
Ratio of air space to grate area	45 per cent.	45 per cent.
Width of air space	⅝ in.	⅝ in.
Tender, type	8-wheeled.	8-wheeled.
Wheels, diameter	33 in.	33 in.
Journals, diameter	5 "	5 "
" length	9 "	9 "
Distance between centers	76 "	76 "
Diameter of center of axle	5⅜ "	5⅝ "
Length of tender over bumper bar	22 ft. 10¾ in.	23 ft. 0 in.
Tank capacity	5,000 gals.	5, 000 gals.
Kind of material	steel.	steel.
Length of tank	19 ft. 6 in.	19 ft. 6 in.
Width of tank	9 " 10 "	9 " 10 "
Height of tank to collar	4 " 6 "	4 " 6 "
Tires, make	Midvale.	Standard Steel Co.
Sight feed lubricator	Michigan triple	Michigan triple.
Front and back couplers	Buckeye short shank.	Buck'ye l'g s'k.
Safety valve	3 in. Coale.	3 in. Hayden.
Sanding devices	Leach "D" 2 double.	Leach "A" double.
Injectors	Two Ohio No. 8.	Two No. 9 Ohio.
Air brake equipment	New York.	New York.
Tender brake beam	Monarch.	Simplex.
" " shoe	Brooks.	A. L. Co.'s.
Steam gauges	Ashcroft.	Star.

General Results.

Boiler Performance.

		No. 80.	No. 80.
1.	Number of trial	2	3
2.	Date of trials	5–3–'04	5–6–'04
3.	Duration of trials, total, hours	2.96	2.64
4.	" " running time	2.58	2.36
5.	" " steaming time	2.31	2.09
6.	Number of stops	13	11
7.	Kind of fuel	Hocking.	Hocking
8.	State of weather	Clear.	Clear.
9.	Direction of wind	E. & S.	E. & S.
10.	Velocity of wind, miles per hour	8	6

Average Pressures.

			No. 80.	No. 80.
11.	Steam pressure at dome	lb.	166	157
12.	Barometer	in.	28.07	29.21
13.	Absolute steam pressure	lb.	180	171
14.	Force of draft, front end	in.	2.05	2.31
15.	" " fire-box	"	1.19	1.5
16.	" " ash-pan	"	.11	.1

Average Temperatures.

17.	External air	deg. F.	65	75
18.	Escaping gases	"	728.3	753.
19.	Feed water in tank	"	58.	62.

Fuel.

20.	Size		Lump.	Lump.
21.	Thickness of fire	in.	11–14	11–14
22.	Weight of coal fired	lb.	7,568	7,508
23.	" " before start	"	1,164	1,067
24.	" " during run	"	6,404	6,441
26.	Percentage of moisture in coal by analysis		7.20%	7.28%
26.	Weight of dry coal fired	lb.	5,943	5,972
27.	" refuse in pan	"	715	769
28.	Percentage of refuse in pan to coal		9.45%	10.22%
29.	Combustible in refuse	lb.	309	371
30.	Total combustible minus comb. in pan	"	6,131	5,786
31.	" ash by analysis	"	583	655
32.	Weight of ash passing flues	"	177	257
33.	Total refuse passing flues	"	730	959
34.	Percentage of ash lost to total ash		30.4%	39.25%
35.	" refuse through stack to coal	"	9.65	12.78
36.	Equivalent coal actually burned	lb.	5,743	5,566
37.	Net dry coal burned	"	5,320	5,160
38.	" combustible burned	"	4,890	4,565

Fuel Analysis—Proximate Analysis.

39.	Fixed carbon	per cent.	49.24	49.70
40.	Volatile matter	"	35.86	34.30
41.	Moisture	"	7.20	7.28
42.	Ash	"	7.70	8.72

Ultimate Analysis.

43.	Carbon	per cent.	68.9	68.58
44.	Hydrogen	"	5.45	5.45
45.	Oxygen	"	16.	15.
46.	Nitrogen	"	1.2	1.2
47.	Sulphur	"	.75	1.05
48.	Ash	"	7.70	8.72

Analysis of Pan Refuse.

		No. 80.	No. 80.
49.	Combustible.....per cent.	43.28	48.18
50.	Ash..... "	56.72	51.82

Analysis of Stack Refuse.

51.	Combustible.....per cent	75.77	73.20
52.	Ash..... "	24.23	26.80

Fuel per Hour Steaming Time.

53.	Actual coal fired ..... lb.	2,772	3,081
54.	Equivalent coal burned ..... "	2,486	2,663
55.	Combustible burned..... "	2,112	2,187
56.	Actual coal fired per square foot grate ..... "	95.6	106.2
57.	Equivalent coal burned per square foot grate	85.7	91.8
58.	Combustible burned per square foot grate.....	72.7	75.4
59.	" " " " " heating surface.	1.08	1.12

Calorific Value of Fuel.

60.	Calorific value per pound of actual coal by Mahler Calorimeter 12,135		12,184
61.	Calorific value per pound of dry coal.....13,080		13,120
62.	" " " " " combustible 14,265		14,384
63.	" " " " " stack refuse 10,750		10,390

Quality of Steam.

64.	Quality of steam at dome (dry steam-unity)	.9774	.983
65.	" correction	.9843	.9875

Water.

66.	Actual weight of water fed to boiler .....lb.	38,580	40,785
67.	Equivalent weight of water actually evaporated into dry steam ..... lb.	37,963	40,275
68.	Factor of evaporation	1.21	1.207
69.	Equivalent water evaporated into dry steam from and at 212 degrees 45,935		48,612

Water per Hour Steaming Time.

70.	Equivalent evaporation per hour from and at 212 degrees.....15,500		18,400
71.	Equivalent evaporation per hour from and at 212 degrees per square foot heating surface	7.97	9.45
72.	Horse-power developed by boiler (34.5 lb. rating)..	449	533

Economic Results.

	No. 80.	No. 80.
73. Water apparently evaporated under actual conditions per pound of coal as fired	6.02	6.33
74. Equivalent evaporation from and at 212 degrees per pound of actual coal fired	7.18	7.55
75. Equivalent evaporation from and at 212 degrees per pound of dry coal fired	7.73	8.14
76. Equivalent evaporation from and at 212 degrees per pound of combustible, minus pan combustible	8.85	9.81
77. Equivalent evaporation from and at 212 degrees per pound of equivalent coal burned	8.00	8.73
78. Equivalent evaporation from and at 212 degrees per pound of dry coal burned	8.63	9.42
79. Equivalent evaporation from and at 212 degrees per pound of combustible burned	9.4	10.6

Efficiencies.

80. Efficiency of boiler from combustible, minus pan combustible	60.	64.
81 Efficiency boiler from combustible burned	63.7	69.
82. " of boiler and furnace from coal fired	57.2	59.8

Flue Gas Analysis by Weight.

83. CO_2 = Carbon dioxide	15.41	15.45
84. O = Oxygen	7.25	8.54
85. CO = Carbon monoxide	1.14	.01
86. N = Nitrogen	76.20	76.00
87. Percentage of air excess	46.44	59.09

Heat Balance.

	B. t. u.	%	B.t.u.	%
88. Loss per pound of coal due to products of combustion	1,371	11.3	1,343	11.03
89. Loss per pound of coal due to air excess	539	4.44	683	5.61
90. " " " " " " " latent heat	497	4.09	492	4.04
91. " " " " " " " unburned coal in pan	538	4.44	657	5.40
92. Loss per pound of coal due to unburned coal passing out stack	1,140	9.39	1,320	10.83
93. Loss per pound of coal due to CO	671	5.53	5	.04
94. " " " " " " " radiation, etc.	439	3.61	394	3.24
95. Heat used in evaporation	6,940	57.2	7,290	59.81
96. Total heat supplied	12,135		12,184	

Engine Performance.

	No. 80.	No. 80.
Number of cars in train	4	4
Gross load in tons	143	143
Total miles	123.7	123.7
Total revolutions		37,830
Average speed in miles per hour	47.9	52.4
" revolutions per minute	244	266
Steam pressure by gauge	166	157
Horse-power, right cylinder, head end	148	169
" " " crank end	140	161
" left " head end	157	183
" " " crank end	169	192
Total	614	705
Safety valve, number of times popped	9	3
" " number of seconds on	705	150
" " pounds of steam escaped	1,596	340
Steam lost at calorimeter and aspirator	229	224
Dry steam lost by pop. calorimeters and aspirator	1,825	564
Dry steam used by engine and air pump	36,138	39,711
Total hours throttle open	2.31	2.09
Indicated horse-power, hours	1,420	1,472
Pounds dry steam per indicated horse-power per hour including air pump	25.6	27
Total strokes of air pump	5,010	5,136
Dry steam used by air pump	731	750
Pounds dry steam per indicated horse-power per engine alone	24.94	26.41
Ton miles	17,690	17,690
Pounds coal per ton mile	lb. .362	.364
" steam per ton mile	" 2	2.2

Boiler Performance.

	No. 73.	No 73.
1. Number of trial	4	5
2. Date of trials	5-13-'04	5-21-'04
3. Duration of trials total, hours	3.09	3.02
4. " " running time	2.61	2.65
5. " " steaming time	2.34	2.34
6. Number of stops	13	13
7. Kind of fuel	Hocking.	Hocking.
8. State of weather	Clear / Cloudy	Clear.
9. Direction of wind	S.W.	N.W.
10. Velocity of wind, miles per hour	17	8

Average Pressures.

11. Steam pressure at dome	lb.	174	170
12. Barometer	in.	29.13	29.21

			No. 73.	No. 73.
13.	Absolute steam pressure	lb.	188	184
14.	Force of draft, front end	in.	2.5	3.
15.	" " fire-box	"	1.91	1.8
16.	" " ash-pan	"	.17	.2

Average Temperatures.

17.	External air	deg. F.	68.	69.
18.	Escaping gases	"	778.	775.
19.	Feed water in tank	"	76.	67.

Fuel.

20.	Size		Lump.	Lump.
21.	Thickness of fire	in.	11.14	11.14
22.	Weight of coal fired	lb.	8,197	8,865
23.	" " before start	"	1,746	1,746
24.	" " during run	"	6,451	7,119
25.	Percentage of moisture in coal by analysis		6.84%	6.70%
26.	Weight of dry coal fired	lb.	6,010	6,642
27.	" of refuse in pan	"	952	830
28.	Percentage of refuse in pan to coal		11.62%	9.35%
29.	Combustible in refuse	lb.	396	289
30.	Total combustible minus comb. in pan	"	6,403	7,113
31.	Total ash by analysis	"	838	869
32.	Weight of ash passing flues	"	282	328
33.	Total refuse passing flues	"	874	1,285
34.	Percentage of ash lost to total ash		33.66%	37.8%
35.	" of refuse through stack to coal		10.66%	14.5%
36.	Equivalent coal actually burned	lb.	5,667	5,996
37.	Net dry coal burned	"	5,280	5,585
38.	" combustible burned	"	4,698	5,000

Fuel Analysis—Proximate Analysis.

39.	Fixed carbon	per cent.	48.78	47.84
40.	Volatile matter	"	34.16	35.66
41.	Moisture	"	6.84	6.70
42.	Ash	"	10.22	9.80

Ultimate Analysis.

43.	Carbon	per cent.	69.13	68.08
44.	Hydrogen	"	5.40	5.40
45.	Oxygen	"	13.	14.
46.	Nitrogen	"	1.2	1.2
47.	Sulphur	"	1.05	1.52
48.	Ash	"	10.22	9.80

Analysis of Pan Refuse.

		No. 73.	No. 73.
49.	Combustible.....per cent.	41.62	34.86
50.	Ash..... "	58.38	65.14

Analysis of Stack Refuse.

51.	Combustible.....per cent.	67.68	74.47
52.	Ash..... "	32.32	25.53

Fuel per Hour Steaming Time.

53.	Actual coal fired ..... lb.	2,757	3,042
54.	Equivalent coal burned ..... "	2,421	2,562
55.	Combustable burned..... "	2,007	2.136
56.	Actual coal fire per square foot grate ..... "	89.5	98.7
57.	Equivalent coal burned per square foot grate	78.6	83.2
58.	Combustible burned per square foot grate.....	65.1	69.3
59.	" " " " " heating surface	1.01	1.08

Calorific Value of Fuel.

60.	Calorific value per pound of actual coal by Mahler calorimeter 12,230		12,151
61.	Calorific value per pound of dry coal.....13,130		13,020
62.	" " " " combustible.....14,775		14,560
63.	" " " " stack refuse 9,590		10,550

Quality of Steam.

64.	Quality of steam at dome (dry steam—unity).....	.9824	.9824
65.	Quality correction	.987	.9871

Water.

66.	Actual weight of water fed to boiler ..... lb.	43,940	43,950
67.	Equivalent weight of water actually evaporated into dry steam	43,369	43,383
68.	Factor of evaporation	1.195	1.202
69.	Equivalent water evaporated into dry steam from and at 212 degrees	51,826	52,146

Water per Hour Steaming Time.

70.	Equivalent evaporation per hour from and at 212 degrees.....16,770		17,270
71.	Equivalent evaporation per hour from and at 212 degrees per square foot heating surface	8.49	8.75
72.	Horse-power developed by boiler (34.51 lb. rating).	486	500

Economic Results.

	No. 73.	No. 73.
73. Water apparently evaporated under actual conditions per pound of coal as fired	6.81	6.17
74. Equivalent evaporation from and at 212 degrees per pound of actual coal fired	8.03	7.32
75. Equivalent evaporation from and at 212 degrees per pound of dry coal fired	8.62	7.86
76. Equivalent evaporation from and at 212 degrees per pound of combustible minus pan combustible	10.3	9.13
77. Equivalent evaporation from and at 212 degrees per pound of equivalent coal burned	9.14	8.71
78. Equivalent evaporation from and at 212 degrees per pound of dry coal burned	9.82	9.35
79. Equivalent evaporation from and at 212 degrees per pound of combustible burned	11.1	10.4

Efficiencies.

80. Efficiency of boiler from combustible minus pan combustible	67.5	60.6
81. Efficiency boiler from combustible burned	72.	69.3
82. " of boiler and furnace from coal fired	63.4	58.2

Flue Gas Analysis by Weight.

83. CO_2 = Carbon dioxide	16.53	16.11
84. O = Oxygen	7.4	7.56
85. CO = Carbon monoxide	.02	.28
86. N = Nitrogen	76.05	76.05
87. Percentage of air excess	48.11	49.53

Heat Balance.

	B. t. u.	%	B. t. u.	%
88. Loss per pound of coal due to products of combustion	1,528	12.48	1,419	11.67
89. Loss per pound of coal due to air excess	628	5.13	595	4.9
90. " " " " " latent heat	488	3.99	496	4.08
91. " " " " " unburned coal in pan	660	5.39	439	3.61
92. Loss per pound of coal due to unburned coal passing out stack	1,018	8.32	1,653	13.6
93. Loss per pound of coal due to CO	11	.09	147	1.21
94. " " " " " to radiation, etc.	137	1.12	327	2.66
95. Heat used in evaporation	7,760	63.48	7,075	58.27
96. Total heat supplied	12,230		12,151	

Engine Performance.

		No. 73.	No. 74.
Number of cars in train		4	4
Gross load in tons		143	143
Total miles		123.7	123.7
" revolutions		34,201	
Average speed in miles per hour		47.4	46.6
" revolutions per minute		221	218
Steam pressure by gauge		174	170
Horse-power right cylinder, head end		143	163
" " " crank end		148	167
" left " head end		167	174
" " " crank end		171	169
Total		629	673
Safety valve, number of times popped		29	14
" " number of seconds on		1,630	580
" " pounds of steam escaped		1,596	568
Steam lost at calorimeters, dome and aspirator		227	236
Dry steam lost by pop, calorimeters and aspirator		1,823	804
" " used by engine and air pump		41,544	42,579
Total hours throttle open		2.34	2.34
Indicated horse-power hours		1,472	1,576
Pounds dry steam per indicated horse-power per hour including air pump		28.2	27.2
Total strokes of air pump		5,040	5,408
Dry steam used by air pump		753	790
Pounds dry steam per indicated horse-power per engine alone		27.7	26.5
Ton miles		17,690	17,690
Pounds coal per ton mile	lb.	.364	.402
" steam per ton mile	"	2.31	2.36

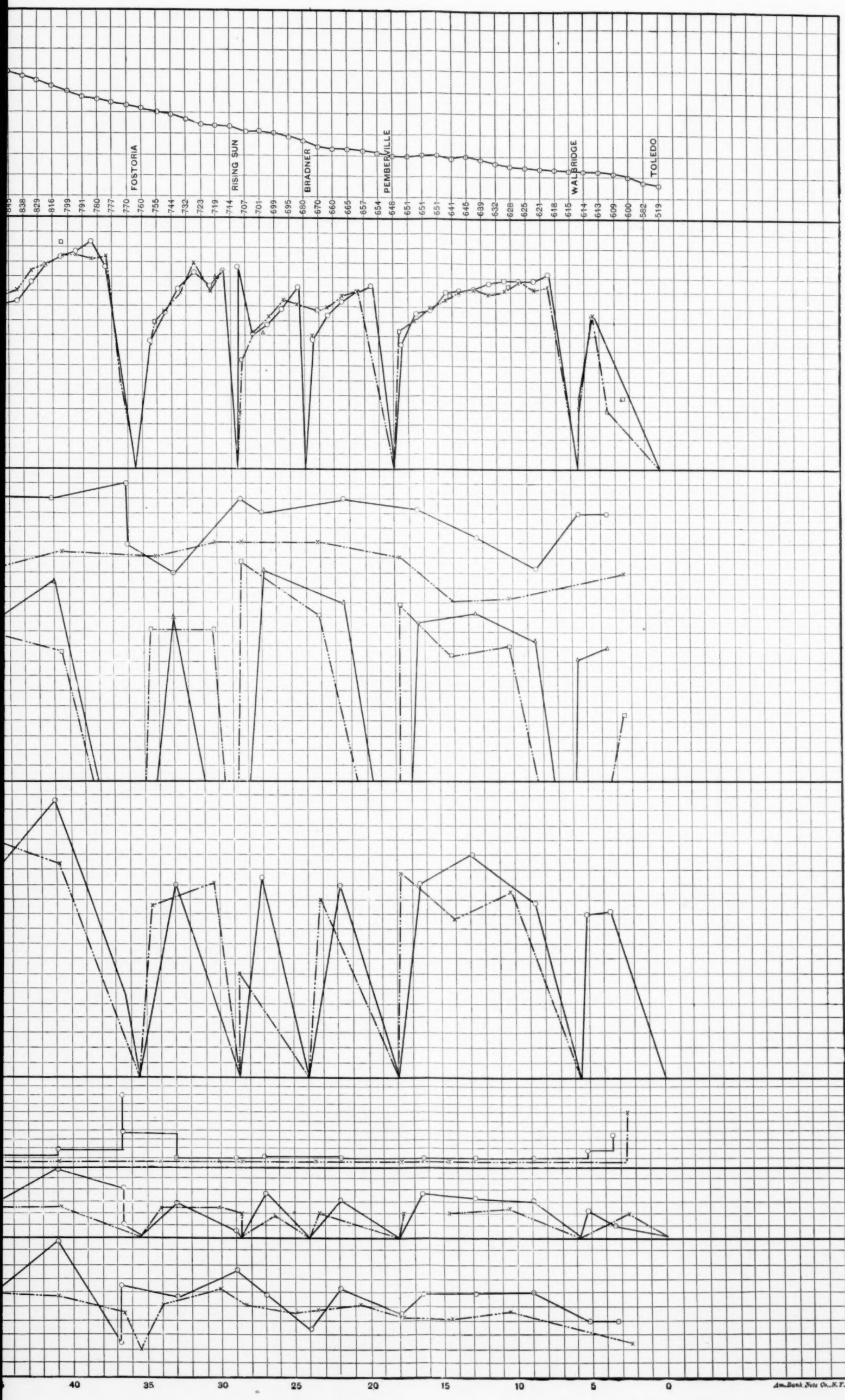
General Results.

13. From the readings taken curves are plotted between mile posts as abscissae and the following: feet elevation of road; miles per hour; steam pressure; initial pressure in cylinder; total horse-power; notches of reverse lever from center; smoke-box draft and smoke-box temperature. There are two sets of curves on each sheet, one set for each engine, thereby showing at once the relative workings of the two at any point in the road.

14. Although the results of four trials are given, five runs were made in all; the results of the first one on number 80 not being given, as it served as a practice trial in many respects, some of the observations not being complete, and then, also, on account of the cool weather it was necessary to have steam on the train,

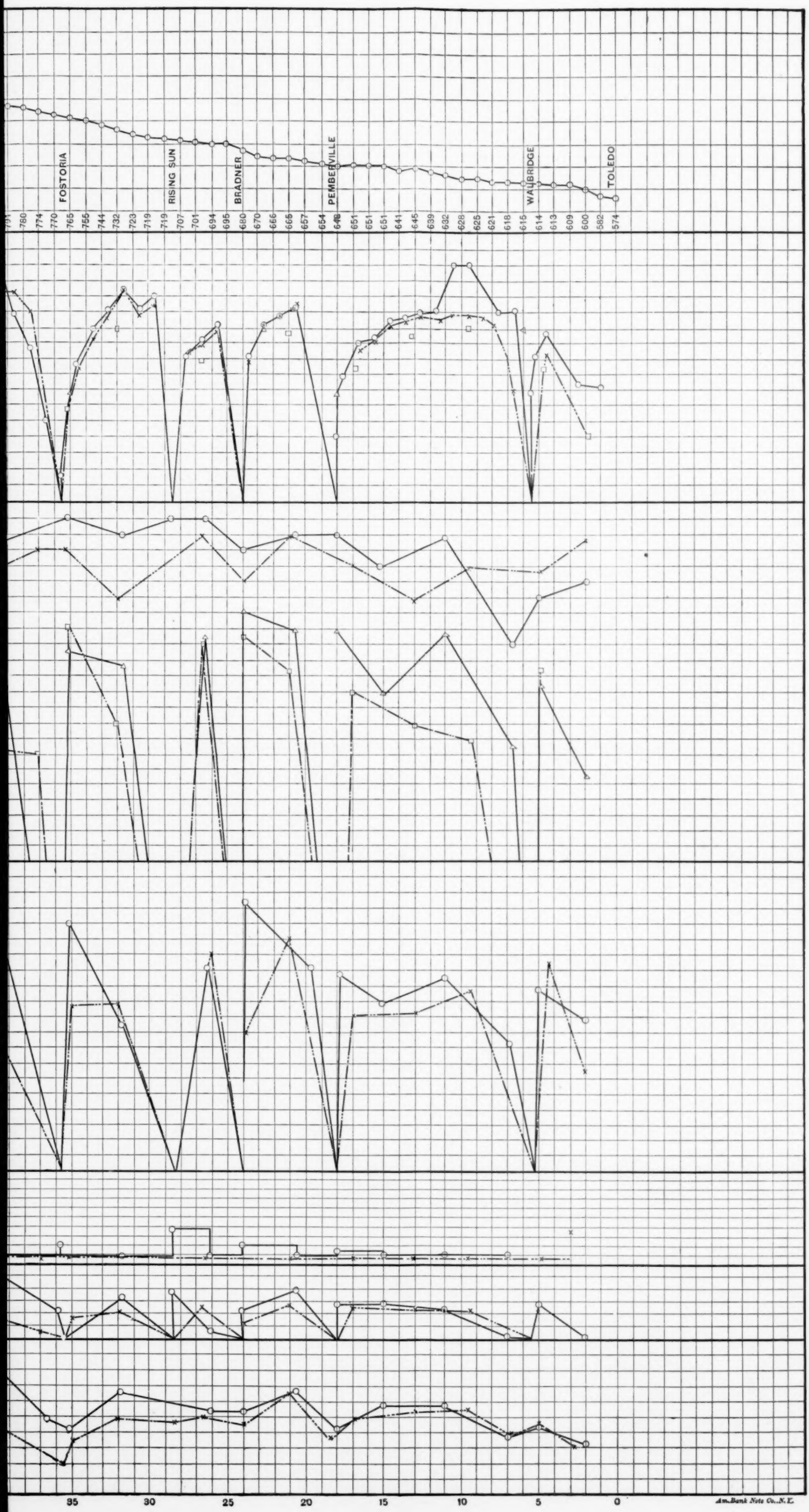
HOCKING VALLEY RY.
FEET ELEVATION ABOVE SEA LEVEL.
Brooks No. 73 Run 5
Brooks No. 80 Run 3
COLUMBUS
DELAWARE
PROSPECT
MARION
UPPER SANDUSKY
CAREY
FOSTORIA
RISING SUN
MILES PER HOUR.
STEAM PRESSURES.
INITIAL PRESSURES.
TOTAL HORSE POWER.
NOTCHES OF R.L. FROM CENTER.
SMOKE BOX DRAFT
SMOKE BOX TEMPERATURES.
MILE POSTS OR MILES FROM TOLEDO.

FIG. 91.


FOSTORIA
RISING SUN
BRADNER
PEMBERVILLE
WALBRIDGE
TOLEDO
40
35
30
25
20
15
10
5
0
Am. Bank Note Co., N. Y.

HOCKING VALLEY RY.
Brooks No. 73 Run 4.
Brooks No. 80 Run 2.
FEET ELEVATION ABOVE SEA LEVEL.
COLUMBUS
DELAWARE
PROSPECT
MARION
UPPER SANDUSKY
CAREY
FOSTORIA
RISING SUN
MILES PER HOUR.
STEAM PRESSURES.
INITIAL PRESSURES.
TOTAL HORSE POWER.
NOTCHES OF R.L. FROM CENTER.
SMOKE BOX DRAFT.
SMOKE BOX TEMPERATURES.
MILE POSTS OR MILES FROM TOLEDO.

FIG. 92.


FOSTORIA
RISING SUN
BRADNER
PEMBERVILLE
WAUBRIDGE
TOLEDO
35
30
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15
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5
0
Am. Bank Note Co., N.Y.

which fact, however, when the following runs were made, gave some figures on the amount of steam required to heat the train, made up of four vestibule cars; that is, one combined baggage and mail car, two coaches and one parlor car.

15. The total water used on this run was about 4,200 pounds more than on the following trials, all conditions being very nearly the same, with the exception of wind and outside temperature, which was 11 miles per hour from the north, with a temperature of 43 degrees F., as against 8 miles per hour from the southeast at a temperature of 65 degrees F. As will be seen, all the general results lie quite closely together, or as close as could be expected, when consideration is taken of the irregularities that are bound to creep in on a regular service test. In comparing runs 2 and 3, in the latter case the speed was much higher and consequently the amount of steam used greater. This was due to lateness in starting, thereby making an average speed in miles per hour of 52.4 and a maximum indicated horse-power at one point of 1,383, yet the maximum speed for No. 80 was on trial No. 2 78 miles per hour. On trials 4 and 5, with engine No. 73, the maximum speed attained was very nearly 82 miles per hour and the maximum indicated horse-power 935, there not being such a variation as in the trials of No. 80 since the times in motion were almost identical. In every case the highest speeds, as shown on the curve, were reached at the same points in the road, between mile posts 90 and 91, also between mile posts 68 and 69.

16. In comparing the several heat balances, although as a whole they are quite similar, the slight variations that do exist may be explained. With trials 2 and 3 the greater loss out of the stack in trial 3 is due to the much higher speed. Trial 2 shows a much greater loss, due to formation of CO. If that factor had been practically eliminated, as in trial No. 3, although the loss due to air excess would probably have increased, due to more free oxygen in the escaping gases, yet there would be a consequent increase in the boiler efficiency, putting it above that of trial No. 3, on account of the less loss from the stack. Therefore, with the same losses out the stack the efficiencies would be very nearly the same. In trials 4 and 5 the greatest difference is in the loss from the stack, with also about the same difference between the boiler efficiencies, but when the difference in the loss due to unburned coal in the pan is taken into account, the boiler efficiency for some reason on trial 4 is too high. This is accounted for by the fact that

during trial No. 4 the mud ring started to leak and, as this could not very conveniently be gauged, or very accurately estimated, the total amount of fed-water to boiler used in computing the final results, the evaporating and consequently the boiler efficiency would run higher than the true value, with a consequent reduction, as found, in the loss due to radiation, etc.

17. Considering this loss on trial 4 as unreliable and taking the average unaccountable loss of the other trials and comparing with the same losses obtained in the same way on the consolidation freight referred to before, the passenger runs give 3.18 per cent. and 1.55 per cent. for the freight, which figures would not necessarily indicate that the loss in this direction was less in case of the consolidation engines, in view of the way this loss is determined, that is, by difference, thereby containing all errors of observation, sampling, etc., and thereby a small percentage of difference in one large item of the heat balance would make a large percentage in that of the "unaccountable loss and radiation." All the heat balance results obtained thus far seem to indicate that the loss referred to would be about 3 per cent., which is about what might be expected for that type of boiler working under the conditions found.

18. It is also of interest to note that the average loss due to unburned coal out the stack for the two trials with Hocking coal on the freight engine was 14.48 per cent., while for the passenger engine, with the same kind of coal, the average of the four trials is 10.53 per cent.

DISCUSSION.

Mr. H. de B. Parsons.—When data are carefully taken from an engine in actual service, it is well to record the facts for the benefit of others. The records here given were carefully obtained by the speaker during October, 1900. The locomotive tested was run over the West Shore and the New York, Ontario and Western Railroads.

The locomotive was built in 1899, and was put in good order prior to the trials. The coal was weighed and none of the coal was used for banking fires. The water was measured by the contents of the tank in the tender, which tank had a water-gauge column located on each side opposite to its center of gravity. The scales attached to the gauge columns were carefully calibrated.

The readings were taken simultaneously, and the average taken to determine the true water-level.

The general data of the engine is given in the accompanying table; as well as the record of the runs on the southern division—between Weehawken and Middletown—and the middle division—between Middletown and Norwich.

The engine was run entirely in passenger service.

The scheduled speed, including stops, was, on the southern division, 31.28 miles per hour; and on the middle division 32.88 miles northbound and 34.03 miles southbound.

The coal was of poor quality, bituminous, run-of-mine, from Coaldale, Pa.

The grades were very heavy between Cornwall and Norwich.

GENERAL DATA OF ENGINE.

N. Y., O. & W. No. 21.

Number of driving wheels	4
Diameter of " " inches	68
High-pressure cylinder, diameter—inches	18
Stroke, inches	28
Steam pressure, lb	180
Number of tubes	274
Diameter of tubes, inches	2
Length of tubes, ft	11
Grate surface, sq. ft	24
Heating surface, sq. ft	1,739
Weight on drivers, lb	89,700
Ratio heating surface to grate surface	72.5

RECORDS OF RUNS ON SOUTHERN DIVISION.

Weehawken to Middletown—Distance, 78.2 Miles.

O. & W. LOCOMOTIVE No. 21.

Date, October, 1900	8	9	9	10	10	11
Direction of running	South.	North.	South.	North.	South.	North.
Weight of train excl. of locomotive and tender, tons	218	191	219	219	226	225
Ton-miles	17,040	14,930	17,110	17,110	17,670	17,600
Coal burned, pounds	9,660		9,660		9,050	
Water used, "	30,460	34,800	26,300	37,460	27,285	38,490
Average steam while running	168	168	169	171	172	169
Number times injector started	9	12	10	10	7	14
Number of stops	22	17	15	16	16	17
Time lost by detentions, minutes	10	7½	4	4½	4½	5
Time made up, minutes	12½	12	6	10	3½	6
Minutes late, starting	2	5	2	9	0	5½
" " arriving	0	½	0	3½	1	4½
Coal used per ton-mile	0.302		0.263		0.254	
Water " " "	1.790	2.330	1.085	2.198	1.693	2.190

RECORD OF RUNS ON MIDDLE DIVISION.

Middletown to Norwich—Distance, 146.9 Miles.

O. & W. LOCOMOTIVE No. 21.

Date, October, 1900	11	12	15	18	19	20
Direction of running	North.	South.	North.	South.	North.	South.
Weight of train exclusive of locomotive and tender, tons	239	180	201	185	248	202
Ton-miles	35,100	26,440	29,530	27,170	36,430	29,670
Coal burned, pounds	*15,200	10,042	9,560	9,400	12,350	10,880
Water used, pounds	74,350	55,850	57,910	56,280	71,340	62,485
Average steam while running	164	173	174	174	169	171
Number times injector started	14	13	16	11	16	13
Number of stops	7	12	6	13	7	10
Time lost by detentions, minutes	12	16	36	17	9	17
Time made up, minutes	1	16	21	19	6	8
Minutes late, starting	10	0	10	2	9	0
Minutes late, arriving	21	0	26	0	12	8
Coal used per ton-mile	* 0.433	0.380	0.322	0.346	0.339	0.366
Water used per ton-mile	* 2.118	2.112	1.961	2.071	1.958	2.106
Apparent rate of evaporation	4.89	5.56	6.09	5.99	5.78	5.74

* Doubtful.

Mr. G. R. Henderson.—I notice one or two things of considerable interest in this. On page 297, article 35, it states that the percentage varied from 9.6 to 12.8. That will answer one of the questions asked a short time ago.

It seems to me one of the principal points of interest is on page 303, the analyses of flue gas in which the proportion of carbon dioxide is shown as 16½ per cent. That is pretty large for a locomotive.

Mr. A. Bement.—I think 16 per cent. CO_2 can be secured from a locomotive fire without difficulty, during a continuous run, provided the amount of coal required is not too great for proper manipulation of the fire. It, of course, requires careful firing. I am unable to estimate the possible maximum CO_2, but consider that it could not be more than 17 per cent. for any considerable interval of time.

In this connection I have come to the conclusion that it will pay to employ two firemen, or an assistant fireman, on some locomotives, because one man cannot properly handle the fire in large engines hauling important trains. One man cannot supply coal almost continuously, shake the grate every five minutes, and attend to some other minor duties; therefore grates are neglected, but if they were properly shaken, it would have an important beneficial effect on efficiency and capacity, which I have found to be true in some experimental work with locomotives.

Doubling the number of firemen is a matter which does not favorably present itself to operating officials; it would, however, be a solution of some of the difficulties experienced with large engines hauling fast or important trains.

Mr. H. H. Supice.—I have heard this question of the use of two firemen on a locomotive discussed more than once, and I have gathered from the opinions of railroad men that there is generally difficulty in getting two men to work in harmony. They either get in each other's way, or come into conflict with each other, each thinking that the other is trying to lay back and let him do all the work, and so there is continual trouble.

Mr. A. A. Cary.—The general trend of this discussion seems to be towards automatic methods of firing the locomotive. For stoking, Mr. Henderson states that he wants a man with more brawn and muscle than head. Such are the qualifications of automatic stokers.

Mr. Henderson.—I would like to say that in the last ten days I saw a test made of an automatic stoker for locomotives, and the actual amount of coal handled was 17,000 pounds per hour.

Mr. Bement.—My attention has been called by Mr. Webber to the fact that the gas analysis in Professor Hitchcock's paper is given in weight instead of volume. Therefore, the CO_2 given is not very high, but the analysis represents average good prevailing practice.

Mr. A. A. Cary.—There seems to be some confusion in the discussion of this paper regarding the fuel-gas analyses, which is due to these analyses being reported by weight instead of by volume, as is the more common practice.

Thus, by referring to page 299 we would find these analyses, if reported by volume, would be:

	1st Column.	*2nd Column.*
CO_2	10.157 per cent.	10.421 per cent.
O	6.829 "	6.040 "
CO	1.227 "	0.011 "
N	81.787 "	81.528 "
Total	100.000 "	100.000 "

And by referring to page 303 we would find the analyses there, if reported by volume, would be:

	1st Column.	*2nd Column.*
CO_2	11.181 per cent.	10.881 per cent.
O	6.986 "	7.127 "
CO	0.022 "	0.301 "
N	81.811 "	81.691 "
Total	100.000 "	100.000 "

Such analyses, if referred to the results obtained in stationary plants, would indicate fairly good furnace conditions.

Mr. Green.—I would like to ask if any railroad company has tried using pulverized bituminous coal? Could that be used?

Mr. Henderson.—I would say that I have investigated that though not in a very thorough way, still I investigated it and was confronted with the fact that the amount of fuel was much less than needed by the locomotive and the manufacturers did not seem to be in shape to supply the quantity of fuel needed by a locomotive.

Mr. A. A. Cary.—I have had some experience with the use of pulverized coal as a fuel.

Under certain severe conditions it is an ideal fuel but those conditions are such as to restrict its use for the boilers of locomotives.

In the first place a more or less expensive equipment is required for pulverizing the coal so that its particles will not vary greatly in size and for bituminous coal carrying from 20 to 30 per cent. of volatile matter, the coal should all be reduced in size so as to pass through a screen of not over 80 mesh.

The less the percentage of volatile matter, the finer the coal should be pulverized, so that if anthracite coal be used, it should all pass through a screen of 200 mesh.

Anthracite is not a desirable coal to burn as a pulverized fuel.

In the next place, to obtain the best results, pulverized coal should be thoroughly dried before it is delivered into the furnace so that its ignition will be almost instantaneous.

If wet fuel is delivered into a hot furnace a delay in ignition is occasioned by: 1st, the time required for the evaporation of its moisture, and, 2nd, by the steam cloud which is formed about each particle which prevents the oxygen in the air from coming in intimate, immediate contact with the fuel, and such delays often cause the fuel to be partially burned outside or beyond the special furnace which must be provided for its combustion and such a condition reduces the furnace efficiency materially.

We thus see that in order to use pulverized fuel in the locomotive, pulverizing and drying plants must be established along the line of travel and special provisions must be made in the tender to receive this finely ground coal and keep it dry until used. As pulverized coal has a great tendency to absorb atmospheric (or other) moisture, this would not be an easy proposition, and if the coal became perceptibly wet it would be apt to clog and accumulate in the feeding or delivering apparatus.

For burning pulverized coal, a special furnace must be provided entirely surrounded with a refractory brick or tile lining—excepting, of course, at the place where the hot furnace gases escape.

With this lining we find another difficulty due to its comparatively short life, even when constructed with what is ordinarily considered very refractory fire brick, to say nothing of the effect of the jarring such a lining will constantly receive in a locomotive.

When using pulverized coal, we realize more nearly the theoretical temperature of combustion of the coal than by any other commercial system of furnace firing, and such high, concentrated temperature in the furnace causes ordinary fire bricks to melt or "shell" rapidly.

With such conditions existing, I am afraid that pulverized coal will not become a common locomotive fuel, although without considering its attending difficulties, it is second only to oil as a fuel for locomotive use.

No. 1069.*

A BAD CASE OF DISCHARGE OF WATER WITH STEAM FROM WATER-TUBE BOILERS.

CAUSE AND REMEDIES.

BY A. BEMENT, CHICAGO, ILL.
(Member of the Society.)

1. This is a particularly interesting case, because the trouble existed on a large scale, and the two engines seriously affected were of some 10,000 horse-power capacity.

2. The methods employed to study the trouble were unusual, the most valuable feature of which was the construction of an experimental boiler, a reproduction of the large boilers on a small scale, and provided with a transparent glass steam drum. By this means information was obtained which could not otherwise have been secured.

3. The most important result of the investigations concerns the performances of water baffles, a device placed in the front end of the steam drums of water-tube boilers, for the purpose of deflecting the water in circulation downward to prevent it being carried out with the steam. It was found that this device as used in the Heine type of boiler which discharged the steam from the front end of the drum, could be greatly improved.

4. Conclusions regarding the utility of separators for removing water from steam is an interesting feature.

5. Effect of drafts of steam caused by the flow to engines at the interval of valve opening, is presented in a graphic manner, explaining why the same type of boilers serving smaller engines were not affected enough for discharge of water to be noticed.

6. The fact was also made evident that the present methods of determining moisture in steam is unreliable, on account of failure

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* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

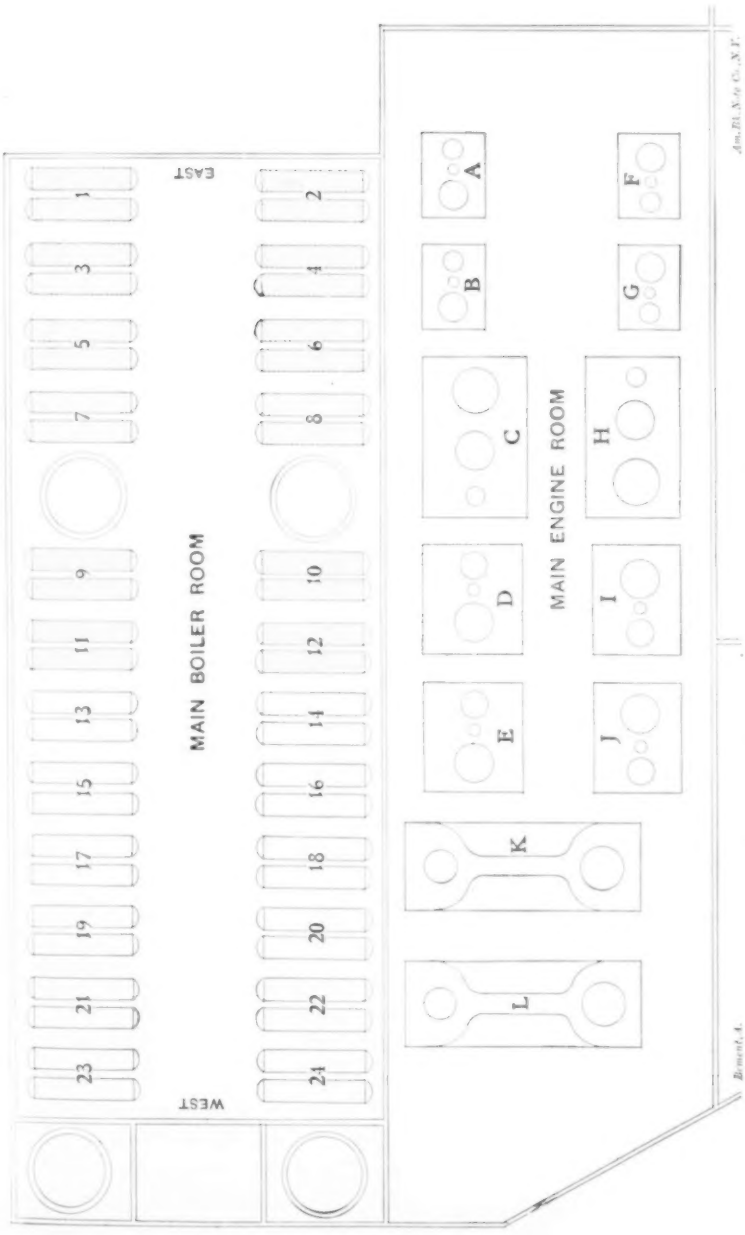


FIG. 93.—PLAN OF MAIN PORTION OF STATION.

to obtain a representative sample. Also that the tendency is for the sample to show most moisture when flow of steam from the boiler is small, and least when the flow is greatest.

Description of Plant.

7. The investigations to be described were made in an electric generating plant of 19,790 kilowatt capacity, of which Fig. 93 is a plan showing the main portion. Boilers Nos. 1 to 24 inclusive are

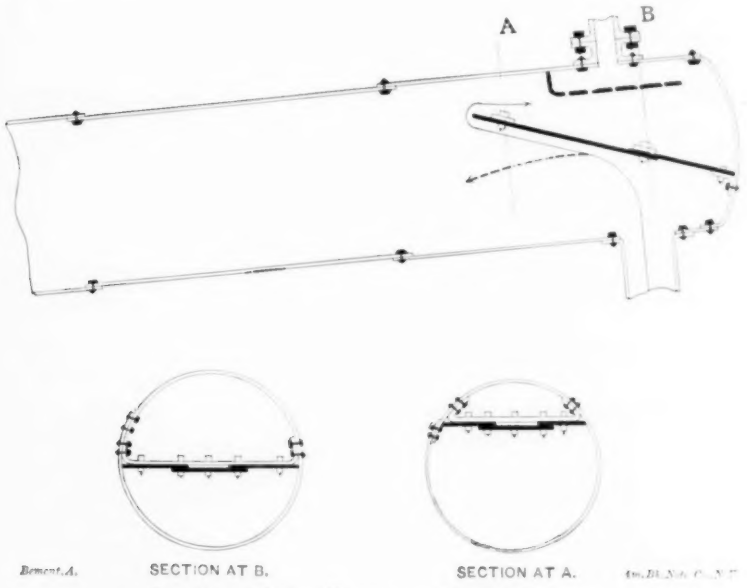


FIG. 94.—WATER BAFFLE IN HEINE BOILERS.

of the same water tube type; Nos. 2, 4, 6, 8, 10, 12, and 14 of 3,800, and the opposite boilers of 4,340 square feet of heating surface each; Nos. 15 to 24 are of 4,800 square feet of heating surface each. All have two steam drums 42 inches diameter by 18 feet long, and are served by a chain grate stoker of an area of 72.2 square feet under each unit.

8. Engines *A* to *J* are vertical triple expansion direct connected units, of 440 each for the smaller and 880 kilowatts each for the larger. *K* is a vertical cross compound Reynolds Corliss engine direct connected to a generator of a maximum kilowatt capacity of 4,500, and *L* is a similar unit of 3,250 kilowatts. Another por-

tion of the plant adjoining the main engine room on the side opposite to the main boiler room, but which is not shown, contains four water tube boilers of another type of 5,000 square feet of heating surface each, and two generating units of 2,500 kilowatts capacity each.

9. The main steam header, 16 inches diameter, is mounted on the boiler room side of the wall separating the boiler and engine room. Steam from the north row of boilers is carried across the room by elevated mains passing above the south row of boilers, from which steam is gathered by branch pipes leading into these mains. Engines *A*, *B*, *C*, *D*, *E*, *K* and *L* take steam directly from the main header, by means of pipes which pass through the wall;

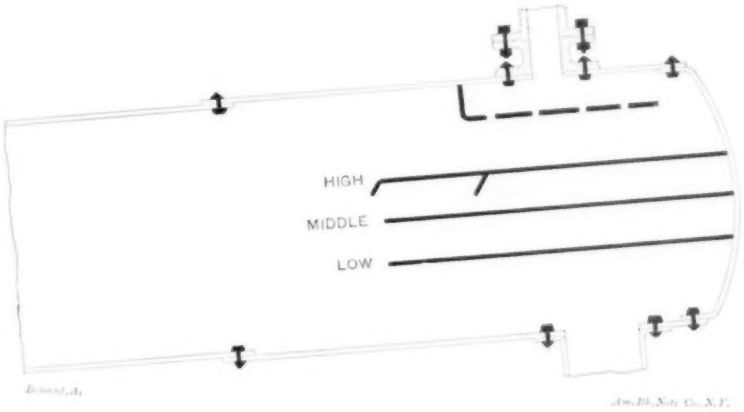


FIG. 95.—STYLE OF BAFFLES.

the diameter of these steam connections are 8 inches for the small engines, 9 inches for the medium, 14 inches for engine *K*, and 12 inches for engine *L*. By means of five tie-over lines leading from the main header, extending down below the floor, and across to the other side, steam is conveyed to a header also below the floor, which serves engines *F*, *G*, *H*, *I* and *J*. The steam system of that portion of the plant not shown, is connected with the header which serves engines *F* to *J*.

10. The main header in the boiler room is in the three portions, one division receiving the steam from boilers Nos. 1 to 8, another from Nos. 9 to 12, and the third from boilers Nos. 13 to 24. Thus there are three individual groups of boilers discharging their steam to individual headers, which are joined by a 9-inch equalizing

pipe. These different boiler groups are shown by differently indicated steam drums. Thus the boilers may be designated as east, middle and west groups.

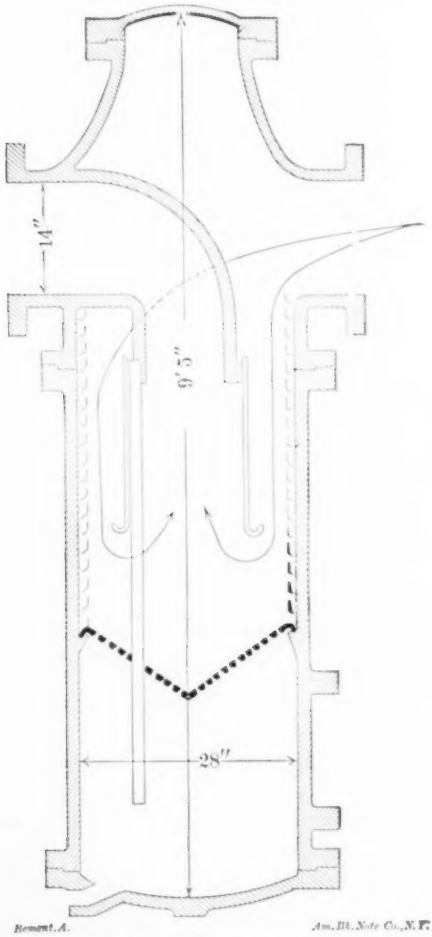


FIG. 96.—SMALL SEPARATOR, ENGINE K.

Manifestation of Trouble.

11. Previous to the occurrence of the troubles to be described, the plant contained engines *A* to *J* and boilers Nos. 1 to 14, and there had been no apparent indication of moisture in the steam being over 1.5 per cent. It was the practice just before the peak

of the load, to fill the boilers to a considerable height, for the purpose of realizing the advantage of a stored water supply during the heavy load, and there was no trouble caused by this practice.

12. In the installation of the west group of units, boilers Nos. 13 and 14 were detached from the middle header, and included in the west group comprising engines *K* and *L*, and boilers Nos. 13 to 24. When this new group was put in service, water was carried

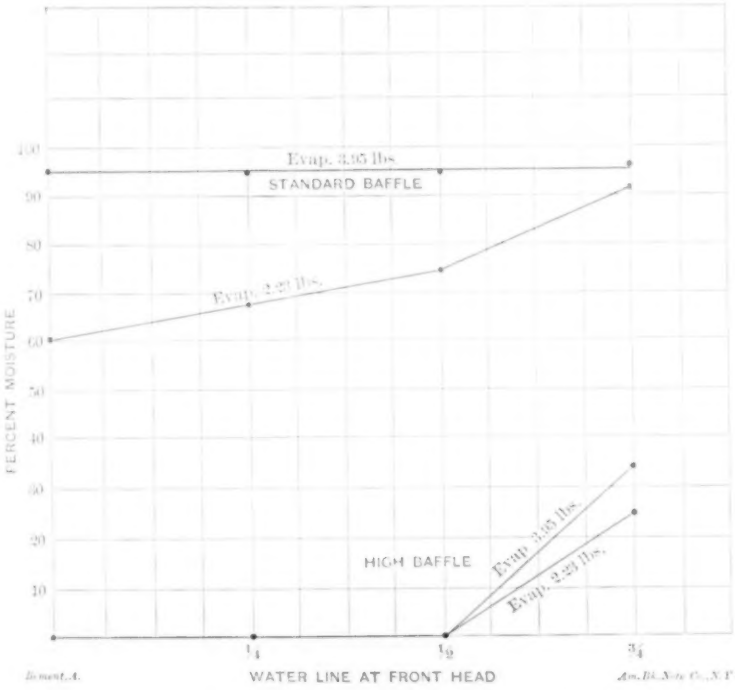


FIG. 97.—RESULTS OF TESTS OF BAFFLES.

to these two engines, resulting in much trouble, as follows: Failure to effect lubrication of valves, resulting in their not closing and the breaking of valve arms. The joints between cylinders and heads is that by a tongue and groove, in which a flexible packing is compressed; this joint failed as often as renewed, and at times water squirted from the joints of the high pressure cylinders. Trouble was always greatest with heavy load on the engines, with either high or low water in the boilers. While there was more trouble with high than with low water, these conditions had less

effect than did the engine load. The amount of water passing the high-pressure cylinders was so great that the drains from the intermediate receiver were not sufficient to remove it; for this reason they were increased in area. These engines have a manhole through the top head of the low-pressure cylinder, closed by an inner and outer cover; before the receiver drains were enlarged, the accumulation of water became so great, on one occasion, to cause a sufficient quantity to enter the low-pressure cylinder

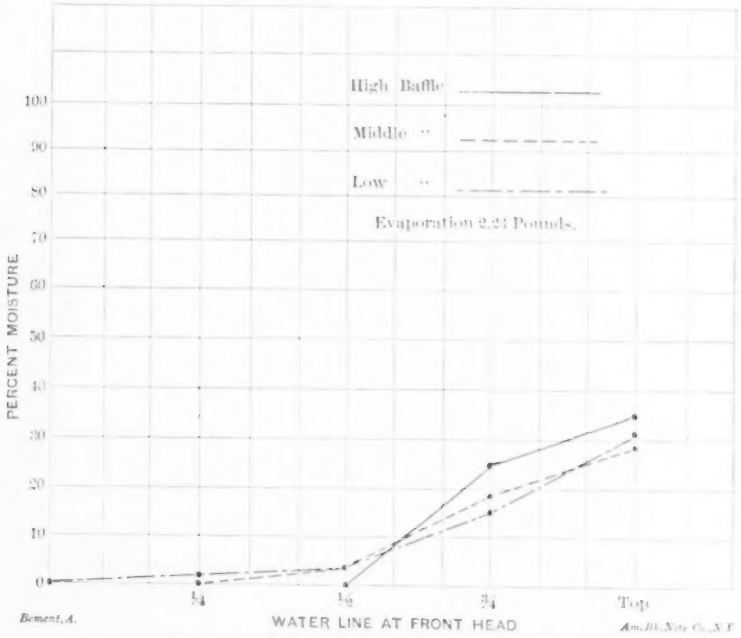


FIG. 98.—RESULTS OF TESTS OF BAFFLES.

of engine *K* to break the bolts holding the inner cover. Owing to failure of high-pressure steam valves to close, the receiver pressure would rise to at least 40 pounds, at which point the relief valve was set; this extreme pressure under the low-pressure piston was sufficient not only to raise the shaft, fly-wheel, generator, etc., but on one occasion caused one of the 4-inch bolts which holds down the cap of the main journal, to be pulled up a sufficient amount to allow the revolving fields of the generator to strike. One of the causes, however, for the failure of this bolt was that it was not properly secured to the foundation.

13. The various troubles above outlined, were of such frequent occurrence as to require these two engines to be taken out of service quite often. There, however, were no such serious troubles as

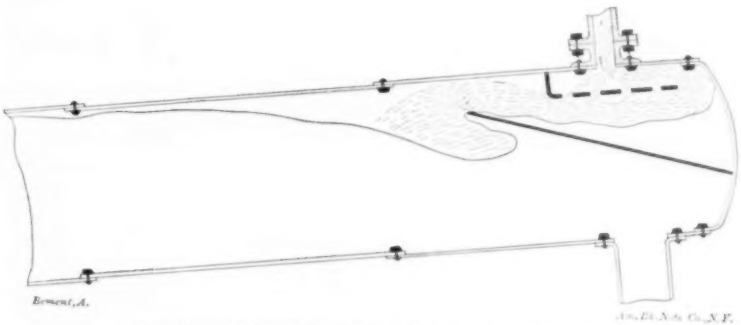


FIG. 99.—HEINE BAFFLE. HIGH WATER LINE.

the breaking of cylinder for example: which could be attributed to water, except in one instance which will be discussed later.

Cause of Trouble.

14. At the above stage of the case the author examined the matter and attributed the cause of trouble to the water baffles in the drums of the boilers, which will be referred to as the standard

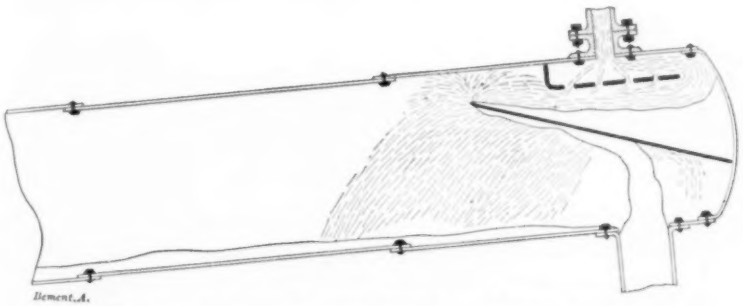


FIG. 100.—HEINE BAFFLE WITH LOW WATER LINE.

form, because it is a standard detail of construction with these boilers.

15. Fig. 94 illustrates the front end of the steam drums of a boiler; as will be observed, the baffle is set at an angle for the purpose of preventing the column of water from the front water leg,

rising directly to the steam outlet. The dotted arrow showing the supposed path of the rising water and the full arrow the path the author considered a portion of the water would take. In

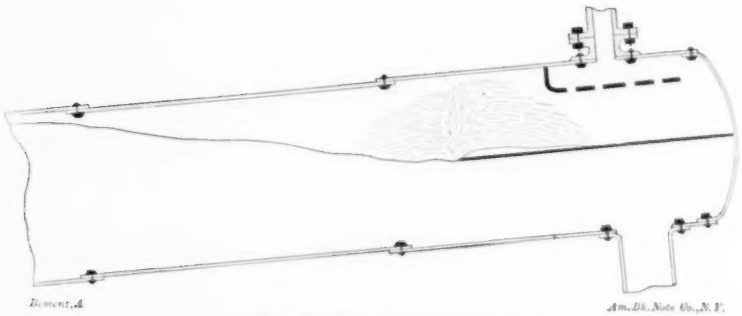


FIG. 101.—MIDDLE FORM OF BAFFLE.

other words, if the flow of steam was rapid enough it would cause some water to be projected to the end of the baffle, from which point it would be carried into the space above, by the high velocity through the contracted passage above the top end of the baffle. As a remedy it was proposed to replace the baffles with that shown in Fig. 95 which is designated as the middle form. It will be observed that it is located at the center of the drum, and affords equal volume of space above and below, and provides

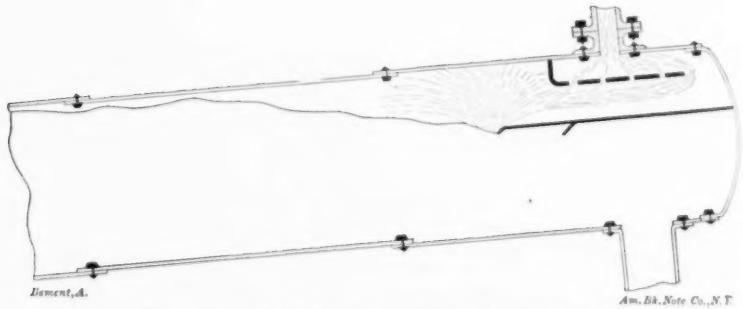


FIG. 102.—HIGH BAFFLE WITH HIGH WATER.

a large area or passage above, and insures a lower velocity of escaping steam. Two other styles of baffles are shown in Fig. 95 which will be mentioned later. The three styles being presented in one figure for simplicity.

First Attempted Remedy, Small Separators.

16. Instead of the proposed baffles being adopted, separators shown by Fig. 96 were placed in the steam pipes to these two engines, that for engine *L* being 7.5 feet in length and 23 inches

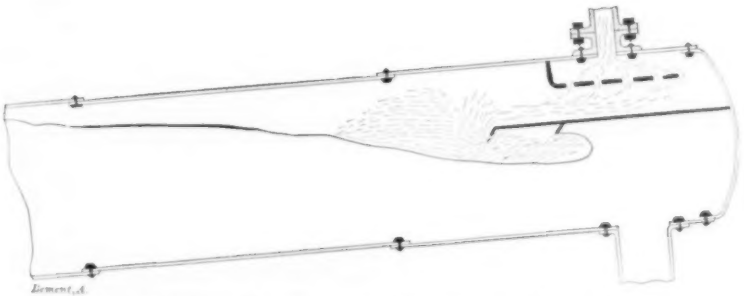


FIG. 103.—HIGH BAFFLE WITH MODERATELY HIGH WATER.

inside measurement, with 12-inch steam openings. The path of steam through these devices is shown by arrows, there being a perforated copper lining as shown. A perforated partition located above the bottom; also two gutters surrounding the central outgoing passage, provided for drainage by a down pipe as shown. The apparent idea of these separators being that the water would flow down the surfaces of the inlet portion of the device and be

FIG. 104.—HIGH BAFFLE WITH NORMAL WATER LINE.

caught by the perforated lining and gutters, and reach the bottom chamber from which it would be removed.

17. When these separators were first put in service, the force of the flowing steam and water tore out the copper linings, which

were about 0.12 inch thick, and they were carried through the high-pressure cylinder into the intermediate receiver, some portions passing through the tubes of the superheater and landing at

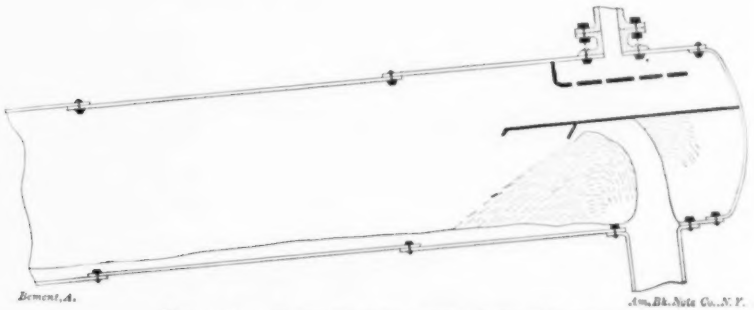


FIG. 105.—HIGH BAFFLE WITH LOW WATER.

the low pressure end. One piece too large to get through became wedged under the throttle valve of one of the engines, and to stop it required that the steam be shut off in the boiler room. Another piece got between the piston and cylinder wall, where it remained till worn away. Steel linings were then inserted, but there was no evidence that these separators removed any of the water which entered with the steam; their presence, however, added to the steam storage space, and for this reason reduced by a small amount the water leaving the boilers for reasons that will be ex-

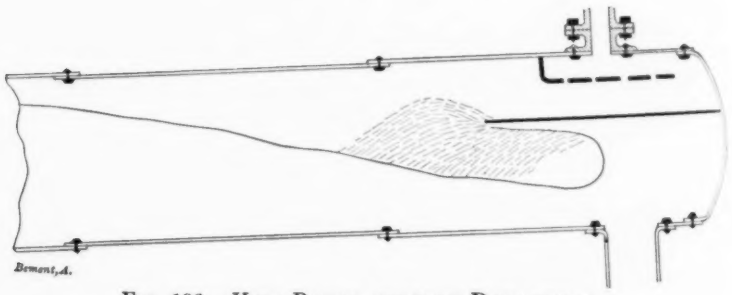


FIG. 106.—HIGH BAFFLE WITHOUT DEFLECTORS.

plained later. Each separator had a Bundy steam trap located in the basement below the engine room, for the purpose of removing the water which it was expected would be collected, but it appeared that all of the water discharged by these traps could be accounted

for by condensation in the lower part of the separator, in the drain pipe and trap. Sometime before these devices were installed, the water line in the boilers was reduced to about one inch in the bottom of the gauge glasses, at which point it was maintained. This together with the increased volume of steam storage space, owing to the addition of the separators, resulted in the trouble from water in the engines being reduced by a noticeable amount, but enough to be of slight benefit only.

Experimental Investigations with Special Boiler.

18. While the problem was at this stage the author began a series of experiments, the results of which follow: During their

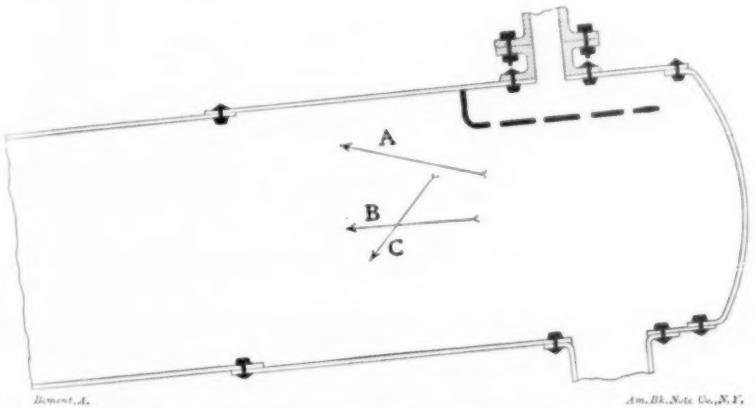


FIG. 107.—ANGLE OF DEFLECTION, DIFFERENT WATER BAFFLES.

progress, however, it was decided to equip these two engines with still larger separators, which has now been done.

19. That the action of different water baffles could be thoroughly studied, a small working model of the boilers was made, and fitted with a glass steam drum so that the entire performance was visible. The heads of the drums were removable; to the front one different forms of baffles were attached for test.

20. The boiler was heated by one blast flame in the tests which gave an evaporation at the rate of 2.23 pounds of water per square foot per hour, and by two blast flames in those with an evaporation at the rate of 3.95 pounds. The air pressure for these flames was supplied by a fan direct connected to a small electric motor, which ran at uniform speed. The gas supply was constant for

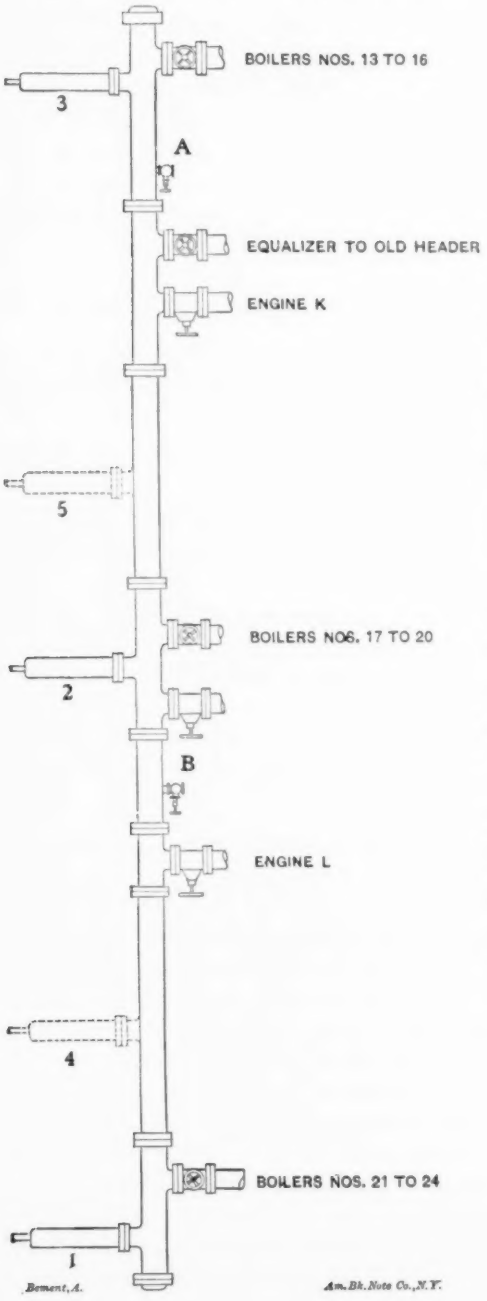


FIG. 108.—STEAM HEADER OF GROUP NO. 3 SERVING ENGINES K AND L.

every experiment. Steam and water was discharged from the boiler by a pipe which entered a glass bottle of about the same cubic capacity as the boiler, passing through a rubber stopper, the steam escaping to the atmosphere by another pipe in the stopper, and the water remaining in the bottle which acted as a separator. The feeding arrangement consisted of an adjustable reservoir, provided with a wide overflow, into which warm water continually ran, the surplus escaping. This reservoir was connected with the front water leg of the boiler by a rubber tube attached to a pipe provided with a feed cock. A glass water gauge was provided at both front and rear of boiler.

21. The experiments consisted in tests of different baffles, at two rates of evaporation. Also four different heights of water line, which were determined by the front gauge glass, according to certain heights on the front head of the steam drum. These water lines shown in Figs. 97 and 98 were those at the beginning of the steaming period. In filling the boiler for each test the reservoir was adjusted to the required height, the feed cock opened until the water in it and the boiler was at the same level; this adjustment of the reservoir was maintained for each water line till all baffles were tested. The initial adjustment of the reservoir was so governed that the water line at rest would produce the desired height when steaming began.

22. The transparent glass bottle used as a separator was not protected against radiation, for this reason some of the entering steam was condensed; to determine the amount of such condensation the boiler was operated for a known period under dry steaming condition; the water in the separator was then measured in cubic centimetres, and divided by the number of minutes of the test. Thus a constant was established for all tests.

23. The performance consisted in the evaporation, or evaporation and discharge of a measured quantity of water, and were not continuous tests at constant water line of regular interval in duration. In each the process was continued till the water line fell to a point equal to two-fifths of the height of the front leg, measured from the bottom of the leg after the fire was shut off. Thus during steaming at the end of the tests the water line was below the upper tubes at the front, but above them in the rear. Measurements of the various quantities above this final water line for the various initial water lines having previously been made, the different quantities for each test was known.

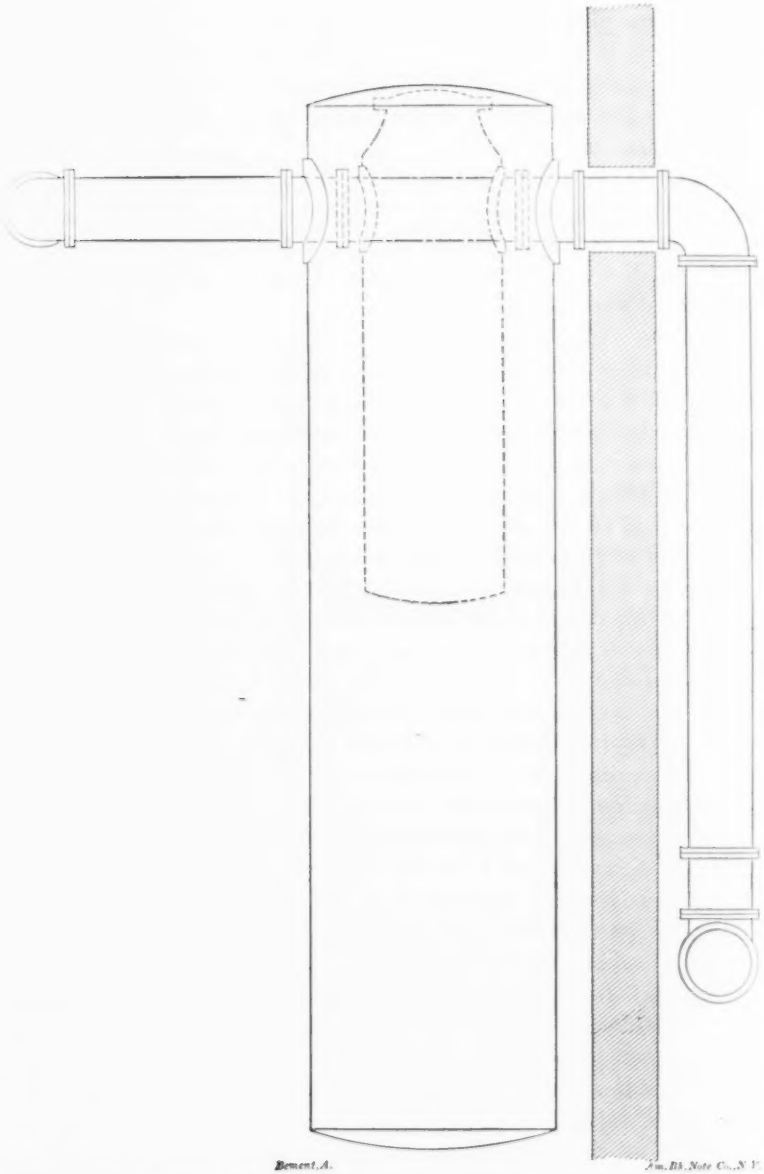


FIG. 109.—ELEVATION SHOWING DIFFERENT STEAM CONNECTIONS TO ENGINES K AND L.

24. In making the tests the initial water line was secured, the feed cock closed and fire applied. At the moment steaming

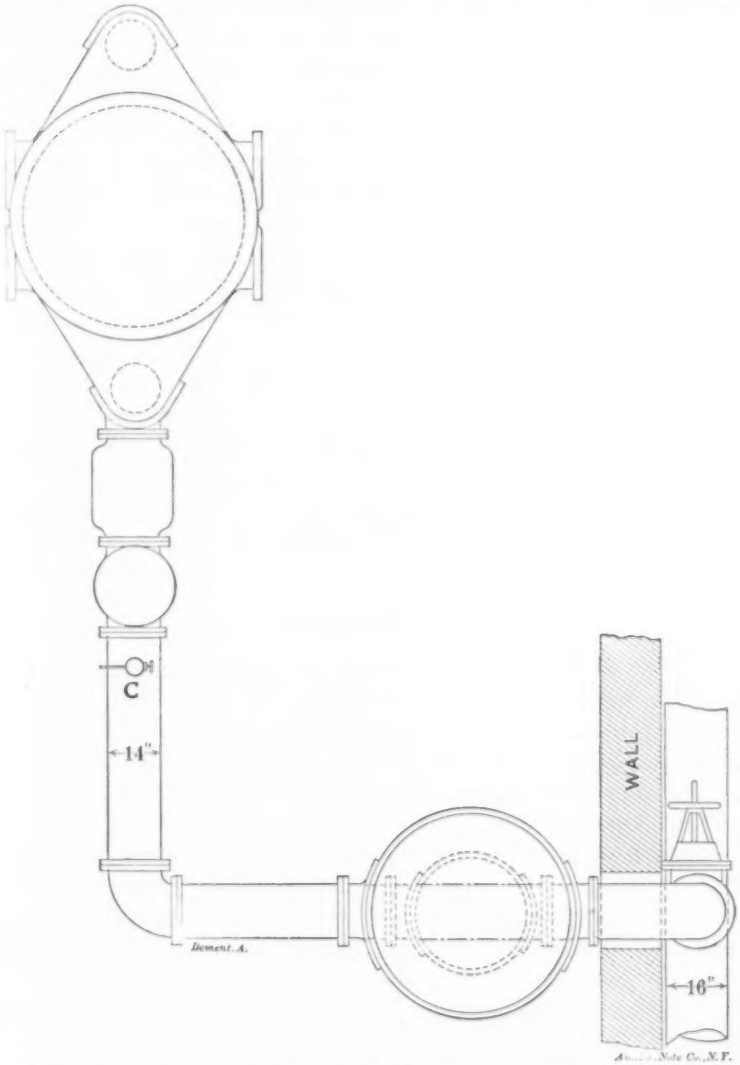


FIG. 110.—PLAN SHOWING DIFFERENT STEAM CONNECTIONS TO ENGINES K AND L.

began time was taken, when the end of the experiment was shown by the height of the water in the front gauge time was taken and

fire shut off. The radiation constant multiplied by the duration of the tests in minutes, gave the total condensation, which substracted from the water in the separator gave the total water carried out with the steam; this referred to the original quantity gave the percentage of moisture in the steam made. The average duration of a test was from 30 seconds to 10 minutes, depending on the baffle and height of initial water line. Altogether 167 regular and a large number of preliminary tests of similar character were made, the results given are based on 93 experiments.

25. The steam pressure with the tests at low rate of evaporation was equal to a water column six inches high, those at higher capacity gave a pressure equal to 10 inches of water; these pressures were due to the resistance of the steam passages which terminated to the atmosphere. The tests were, therefore, made at practically atmospheric pressure, for this reason the size of the steam bubbles were very large, which caused violent ebullition, much more than would be the case with higher pressure. This condition served to develop the relative values of the devices over a very large range, as the ejection of water with the standard form, at the high rate of evaporation was carried very nearly to the maximum.

Results with Experimental Boiler.

26. Fig. 97 is a diagram of results of performance of the standard baffle which is illustrated in Fig. 94, and of the high form provided with deflectors shown in connection with others in Fig. 95. It will be observed that with the high baffle moisture is not carried out till the water line is above the middle of the front head; above this point it increases rapidly. At low rate of evaporation moisture increases with the standard baffle with increased height of water line, but at high rate of evaporation it remains practically constant, showing that if conditions are severe enough nearly all of the water will be ejected regardless of the height of water line. It is, of course, not to be inferred because the standard baffle allowed the escape of 95 per cent. of the water with the model that the large boilers do so. It means, however, under conditions that this device would produce 95 per cent. moisture, the high form of baffle with deflectors would make dry steam, and that this improvement in service would possess enormous margin over any dry steaming requirements.

27. Diagram Fig. 98 gives comparative performance of the

three styles of baffles illustrated in Fig. 95. It will be observed that when the water line was below them, that dry steam was produced; but when it was above, each allowed of the escape of water, and that this amount was largest with those having the smallest space above owing to the consequent higher velocity of escaping steam.

28. The actual performance of the principal baffles, as viewed through the glass walls of the steam drum of the model boiler is shown in Figs. 99 and 106, which are vertical half sections of the drums of the large boilers with different baffles shown therein.

Fig. 99 is the performance of the standard baffle with high water line, and Fig. 100 with low water. The first thing to occur

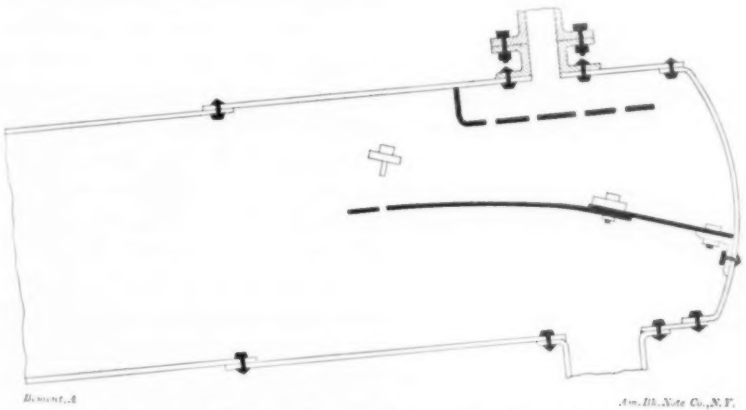


FIG. 111.—END OF BAFFLE PLATES DROPPING DOWN.

when formation of steam began was the projection of a quantity of water up the under surface of the baffle to its end; from this point a portion fell back into the drum, the remainder was carried over the baffle and filled the space above; when this space had become filled, water began to leave with the steam, continuing to do so as shown by Fig. 100 as long as the heat was supplied. The front gauge glass with such conditon would indicate a water level lower than the bottom of the drum, yet the circulation would rise as shown. The water would pile up in the back of the drum when there was a large quantity in the boiler, but when reduced to a small amount, would flow in a thin stream along the bottom.

29. Fig. 101 serves to show the tendency of the action of the middle form of baffle with the surface of the water at the level of its end, about one-half of the spray would be carried onto it, but

owing to the low velocity of the outgoing steam above, would flow back instead of being carried out.

30. Figs. 102 to 105 show the performance of the high baffle provided with two deflectors. This excellent device was the outcome of experiments with the middle form. Without the deflectors as shown in Fig. 106, it was more efficient than the middle form, but like it, the flowing water passed the end where it rose in a spray; for this reason the author added the deflectors, for which there was ample room on account of its height, with the result that the water was projected downward.

31. Fig. 107 illustrates the angles of deflection of the different baffles. *A* represents that of the standard, *B* the middle, and *C* the angle of the deflectors of the high baffle. It will be observed that the difference in angle between *B*, a very efficient device, and *A* an extremely inefficient one, is slight, but that the area above their ends is very different. Also that the effect of the deflectors is to project the water in a direction away from the steam outlet.

32. It was shown in the experiments with the model boiler, that the water rose from the front leg with sufficient force to project it to the top of the standard baffle, and it was desirable to ascertain to what height it would rise if no obstacle intervened. Steam was removed by a pipe extending back in the drum about one-third of the distance from the front end, a glass standpipe was mounted above the regular steam outlet, its upper end being in communication with the steam space in the boiler, so that pressure would equalize above the column of water therein. The baffles were removed so that the rising water from the leg could reach the opening to the standpipe. Fire was applied, and at the moderate rate of evaporation, water rose in a solid column in the standpipe to a height equal to that of the water leg, and at high rate of evaporation to a height nearly twice as great.

Effects of Drafts of Steam on Priming.

33. With an engine cutting off steam for example, at one quarter stroke, it will, during the interval that the valve is open, take steam at a rate four times faster than it is being generated; such steam is in storage in the piping and boiler drums, and with each valve opening the steam pressure will be reduced, with the result that the rate of evaporation in the boilers will be increased for the interval of drop, in proportion to the amount of drop in

pressure occurring in the boilers. The steam pressure employed in the plant in question is 175 pounds above atmosphere, at this temperature a drop of one pound pressure will liberate approximately 0.5 of a heat unit for each pound of water in the boilers, therefore a small fraction of one pound drop in pressure, if occurring in sufficiently short interval would increase the rate of evaporation to an amount enormously greater than the average, and to this is due the trouble from water in engines *K* and *L*, as compared with others in the plant.

34. If steam storage capacity is compared with demand made upon it, useful facts are presented. Referring to Fig. 93, engines *A*, *B* and *C* and boilers Nos. 2, 4, 6 and 8 are designated group No. 1. Engines *D* and *E* and boilers Nos. 10 and 12 compose group No. 2; group No. 3, includes engines *K* and *L* and boilers Nos. 14, 16, 18, 20, 22 and 24. In calculating steam storage volume for these three groups, the cubic contents of the main steam header and engine connections are taken. It is, of course, true that steam is in storage in much other piping, but it cannot respond to the sudden demand to any large extent, because of the brief interval of time. For this reason drop in steam pressure occurs particularly in the boilers composing these groups, owing to their respective engines drawing more directly from them. A large amount of storage space exists in the boiler drums, of course, but this is relatively equal in each group.

TABLE A.

STEAM STORAGE SPACE COMPARED WITH KILOWATT CAPACITY.

Engines and Boilers—Group Number.	Kilowatts.	Relative number of valve openings in given interval.	Cubic feet of storage.	Kilowatts per cubic foot of storage space.	Relative demand made on steam storage space. Group No. 1 the standard.
1	1760	1.73	110	16.00	1.00
2	1760	1.46	76	23.11	2.49
3—Original	7750	1.00	183	42.35	7.92
3—Small separators	7750	1.00	223	34.75	6.50
3—Large separators	7750	1.00	647	11.96	2.23

35. Table *A* shows the relations of the groups in reference to kilowatt output per cubic foot of storage space. The relation in group No. 3 is given as the engines were originally installed

without separators, also with the small separators, and with still larger ones now in use. Engines *A* and *B* run at 150 revolutions per minute, *C* at 90 and *D* and *E* at 110 revolutions, while engines *K* and *L* make 75 revolutions per minute. Therefore these larger engines which make less number of revolutions in a given time, must take proportionately more steam at each valve opening, for this reason draw harder on the stored supply of steam, with the result that drop in pressure in the boilers of group No. 3 is in proportion. The figures in the last column of the table are prepared in accordance with this fact.

36. Calorimetric tests for moisture in steam from boilers of group No. 3, and measurements for drop in pressure at engine *K;* in steam header of group No. 3, also in steam header of group No. 1, were made during the time that the small separators on engine *K* and *L* were in service. The results of these experiments will follow: But attention will be first directed to Figs. 108, 109 and 110 showing the steam header serving these two large engines, with its various connections; also a plan and elevation of the three different conditions of the steam connections to them, showing the large separators in full outline, and the small ones and the original straight steam pipe in dotted outline.

TABLE B.

CHANGE IN STEAM PRESSURE, DUE TO ENGINES TAKING STEAM, AVERAGE CONDITIONS.

Location of measurements.	Variation in pressure.	Drop in pressure
	Lb.	*Lb.*
At the throttle of engine K.....	25.0	16.25
Steam header group No. 3 at A. Fig. 107.....	11.0	5.5
" " " " " " B.....	8.0	4.0
" pipe from boiler No. 14.....	1.0	
" header at engine C. of group No. 1.....	4.0	2.0

37. The pressure measurements given in Table *B* were made with a Crosby steam engine indicator, using a 100-pound spring. Location of these measurements are shown at *A* and *B* of Fig. 108, and at *C* of Fig. 110. The measurement in the steam header of group No. 1, at the outlet to engine *C*, corresponded in location to similar measurements, in the header of group No. 3. The change of pressure in the steam pipe at boiler No. 14, as near as the diagram could be measured, appeared to be one pound, which

could, however, be assumed to be even less. It is impossible, in fact, to measure the drop in pressure in the boiler drums, because there can be no drop, for the reason that the liberation of latent heat results in evaporation of water, and the resulting steam immediately tends to restore the pressure. This action influences the pressure in the steam pipe at the boilers to a large extent, and likewise in much less measure affects the drop in the header. Therefore, there is no drop in pressure in the boilers, but there is increased evaporation. There would, of course, be an extremely small amount of lag in the response of the evaporation in restoring the steam removed, and this if anything is what could be measured in the steam pipe at the boiler.

38. Upon the opening of a steam valve, pressure would drop, and steam flow to the cylinder at considerable velocity; at valve closure this flow would suddenly be arrested, resulting in the inertia raising the pressure in the piping, which would, of course, fall to normal before the valve opened again. Therefore, the variation in pressures given in Table *B* are both drop and surge, between which the indicator does not distinguish. For this reason a Bristol recording gauge was attached to the same point with the indicator, the change of pressure making a record on the moving chart. When this was obtained the gauge cock was throttled, so that the pressure in the gauge was constant; then the resulting record gave evidence of how much of the change was due to drop below normal and that due to surge above normal. Thus figures for the last column of Table *B* were obtained.

39. A very interesting proof that there is a surge, was afforded by experiments with the steam calorimeter, which was attached at *C* in Fig. 110. This location was as much as eight feet away from the point where the cylinder oil entered, yet oil was carried back and entered the calorimeter, resting on the water as shown in the glass gauge.

40. At the opening of steam valves, pressure drops and rate of evaporation in the boilers increases. At valve closure the flow is arrested, resulting in pressure in the piping being raised above normal, causing a return surge to the boilers raising the pressure therein and stopping evaporation entirely for a brief interval.

41. Thus the performance in the boilers is divided into three periods; a very high peak of evaporation, followed by a brief interval of no evaporation at all, and then a longer period of evaporation at normal rate, the average rate being in proportion to

the absorption of heat by the boilers. It is these peaks of evaporation which cause the water, or most of it, to be carried over the standard baffles, thence out with the steam to the engines.

Effect of Water Hammer on Baffles.

42. The foregoing discussion of the performance of the process of evaporation while being correct, offers no corroborative evidence. It is, therefore, in order to submit proof of the fact. If the engines and boilers of group No. 3 were a unit, it would be possible to calculate the effect of drop in pressure, and express the

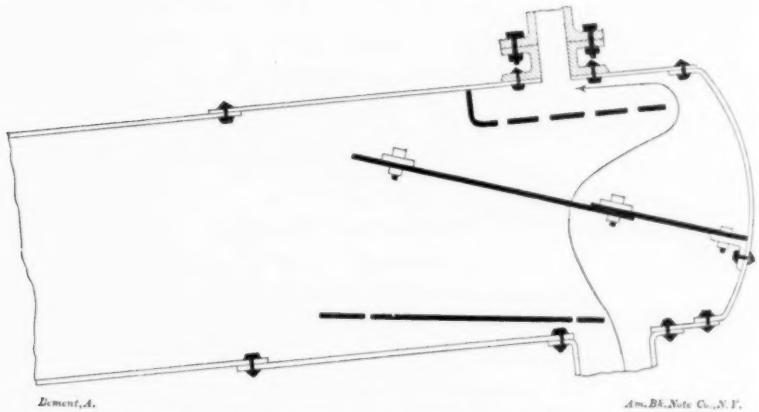


FIG. 112.—ELEVATION, PLATE COMPLETELY DETACHED.

result in rate of evaporation, but this group is connected, not only with an equal number of boilers on the opposite side of the room, but with the other portion of the plant by means of a 9-inch equalizer. For this reason stored steam in these portions has an unknown influence on the problem. That such influence is not large, however, will be shown. Reference to Fig. 94 will illustrate the construction of the standard water baffles as furnished in boilers Nos. 15 to 24. As observed they are composed of six pieces of steel plate $\frac{3}{16}$-inch thick, bolted to the two traverse braces, the baffle being made up in the width of three pieces, and in length of two pieces, those in contact with the head of the drum being attached to it.

43. With every one of these baffles in the drums of the boilers of group No. 3 failure has occurred by nuts becoming unscrewed and dropping off, causing the plates to become detached so that to

a greater or less extent every baffle has come to pieces. Examples of these failures are illustrated in Figs. 111 to 114.

44. Fig. 111 is an example of the most common failure, and consisted in the nuts coming off and allowing the upper end of the plates to drop down. There is no evidence that this kind of failure increased the moisture in the steam; it probably had an opposite effect, because in a measure it acted in a similar manner to the middle form of baffle tested in the model boiler.

45. Fig. 112 shows the back center plate completely detached, and resting on the bottom of the drum. Fig. 113 is a plan view, showing the vacancy with plate below. The location of this de-

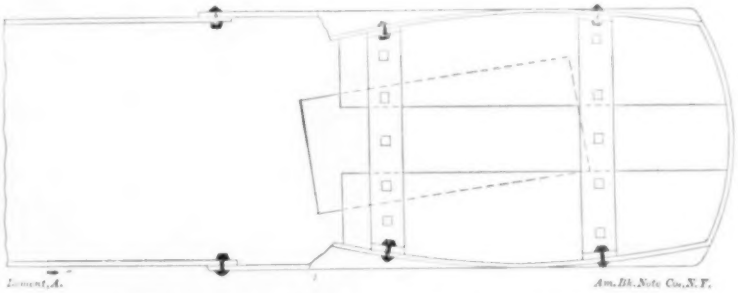


Fig. 113.—Plan, Plate Completely Detached.

tached plate when the boiler was opened is that indicated, which was not necessarily the same when the boiler was in service.

46. Fig. 114 illustrates a very bad failure, as it allowed water to rise directly above the baffle, filling the space so that the steam passing over the end carried it out in large mass. This failure occurred in boiler No.16, and was immediately followed by a great rush of water to engine *K*, which squirted out of the high-pressure cylinder head joints, raised the shaft, stretched the studs and destroyed the gasket under the cylinder head. The next Sunday when permanent repairs were attempted it was discovered that the cylinder was cracked in the fillet of the flange. This was the most serious rush of the water entering either of these engines; it produced a tremendous upheaval which shook the building, with only the damage described, which serves to indicate that a large amount of water may pass through an engine without causing much damage.

47. The baffles in other of the 18 boilers under discussion in

this plant are secured with bolts just the same as are these six, but have never unscrewed and released a plate, although some of them have been in service a period four times as long as those of group No. 3. With the boilers on the opposite side of the room there has been no trouble of this character, although made and installed at the same time, and discharging to the same steam header. These failures can only be accounted for by the peaks of evaporation, the force of the rising water unscrewing the nuts from the bolts. Recurrence of these failures was prevented by riveting the end of the bolts over the nuts.

48. Another exceptionally strong proof of the occurrence of

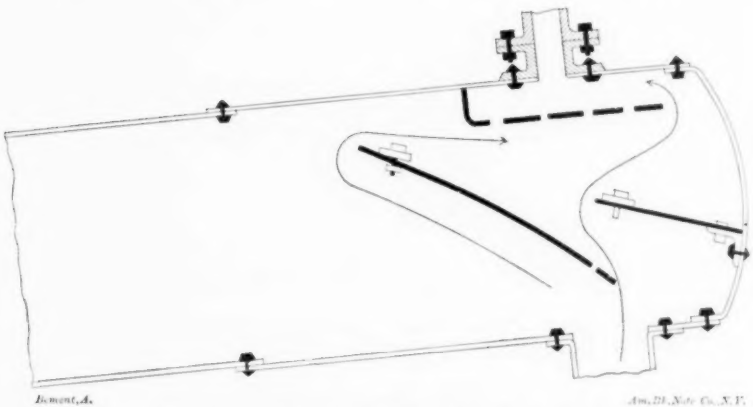


FIG. 114.—EXCEPTIONALLY DANGEROUS FAILURE IN BOILER NO. 16.

these peaks of evaporation is afforded by Fig. 115, showing the old style of baffle in boiler No. 14, which is composed of three pieces of plate of full length, but not attached to the head of the drum, being loose as shown at *A*. With each opening of high pressure steam valves of engine *K* it strikes a blow against the head at *A*, but with engines *K* and *L* out of service and the boiler at maximum capacity there are no blows. Likewise similar baffles in boiler No. 12 of group No. 2, where the steam storage space is not large, strike a similar blow but not near so hard. But with other similar baffles in other portions of the plant there is no sound. The behavior of these baffles not only show that drop in pressure has a serious effect on the boilers of group No. 3, but that it does not extend to any great extent to those on the other side of the room.

49. If it is assumed that there was no hot water in the boilers, but the storage space filled with steam, then opening of a steam valve would cause a reduction of pressure. With the water present there is no actual drop in the boiler drums, but there occurs what may be termed a theoretical drop, which, if being known makes it possible to calculate the rate of evaporation. But for reasons before mentioned, this is not known. Therefore it is assumed that the average theoretical drop in the boilers of this group was 0.15 of a pound, which occurring for a period of 0.1 of a second, corresponding to the valve opening at 75 revolutions per minute, at one quarter cut-off, would result in the evaporation

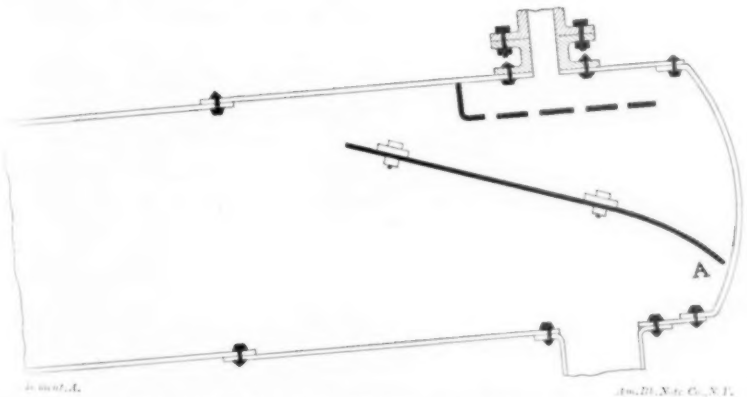


FIG. 115.—OLD STYLE BAFFLE. BOILER NO. 14.

of 2.3292 pounds of water during this interval. At the rate of 23.292 pounds per second, or 83,851 pounds per hour, equal to 2,430 horse-power from one boiler. The contents of a boiler being 30,000 pounds of water, and the liberation of heat equal to 0.5 British thermal unit from each pound for a drop of one pound pressure. At the beginning of valve opening the flow of steam is slow reaching maximum at closure, when the return surge to the boilers raises the pressure and stops evaporation entirely. Fig. 116 is a graphic illustration of this performance, wherein the peaks are reduced to a mere fraction of 0.1 of a second.

Calorimetric Determinations.

50. The usual method employed to determine whether a boiler is discharging wet steam is to test it with a calorimeter. One

hundred and forty-three such tests were made on boilers of group No. 3, the results of which are most interesting and prove the utter fallacy of this business as far as getting an average sample is concerned. It is the author's opinion that the water in a vertical pipe will flow in contact with the walls; for this reason a sample taken from the center of the pipe cannot show an average percentage of water. For this reason the sampling pipes which were duplicates of each other, were perforated for their whole length, so that if possible the sample would be gathered uniformly from the center to surface of the pipe, with the idea of securing any water travelling along the pipe. The 0.5 inch pipes were each bushed with a 0.25 inch pipe, and twenty $\frac{3}{16}$ inch holes drilled through each, and the inner ends of the sampling pipes were plugged. A Carpenter separating calorimeter was used.

51. It is a fact that the harder a boiler is worked the more water it will discharge with the steam, but the most significant showing of the moisture determination was the presentation of the results as being just the opposite. For example tests were made on boiler No. 22 under different conditions with the following results:

Conditions.	Range of moisture per cent.
Engine L only in service, boiler at small capacity.....	11.56 to 17.72
" " " " " " " large "	7.14 to 10.96
Engines L and K in service taking steam at the same interval, and boiler at full capacity	1.66 to 2 11

52. Tests were made under all attainable conditions except that of high water, which was not advisable to attempt. Arrangements were made that the relative crank positions could be made the same, ensuring that these two engines took steam simultaneously. The extreme range of all samples, from all boilers, and conditions, was from 1.37 to 23.0 per cent. moisture. The higher determinations were from boiler No. 18 when the baffle was in the condition shown in Figs. 112 and 113, which will be discussed later. Moisture with everything in normal condition ranged from 1.37 to 17.72 per cent.

53. The probable explanation for the various samples obtained from boiler No. 22 for example, under the different conditions of operation, is a difference in velocity of steam flowing from the boiler. For illustration the outlet from the calorimeter does not exceed $\frac{1}{32}$ of an inch in diameter, while the inlet at the sam-

pling pipe is many times greater. For this reason the velocity of flow of the steam into it is not high, therefore when steam flows from the boiler at high velocity the water is carried past the sampling pipe, but when flowing from the boiler at low velocity, the calorimeter gathers more water, so that when the sample showed 17 per cent., for example, it contained a greater percentage of water than the steam leaving the boiler. Therefore, de-

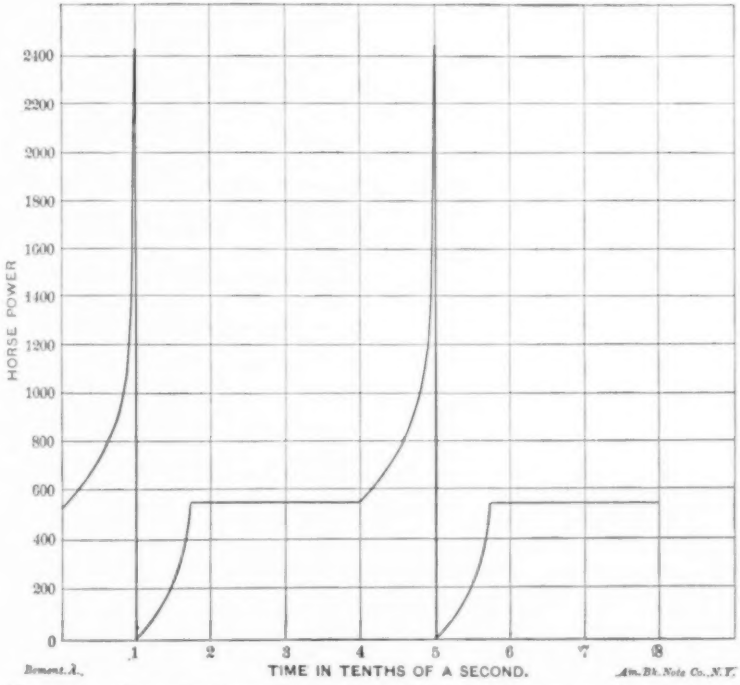


FIG. 116.—RATE OF EVAPORATION ONE BOILER, BOILERS OF GROUP NO. 3 DURING ONE REVOLUTION OF ENGINES K AND L.

pending on conditions, the sample either contained too much or too little water.

54. During these moisture determinations all of the boilers of this third group were equipped with the standard baffle, except No. 20, which was fitted with the middle form shown in Fig. 95. The same tests were applied to it as to No. 22, except that the water line was as much as 6 inches higher. The comparison between the baffles is—

Baffles.	Range of moisture per cent.
Standard baffle, boiler No. 22	1.66 to 17.72
Middle " " " 20	1.51 to 2.11

55. The tests on boiler No. 20 with the middle form of baffle would indicate that it produced much dryer steam than the standard, as proven by the tests in the experimental boiler. But with it most moisture was present in the sample at low capacity, but as the condition which produced 2.11 per cent. with this baffle, made 17.72 per cent. with the other may be taken to mean that it produced dryer steam, as the experimental boiler showed it should do. Yet this 2.11 per cent. moisture was greater than an average sample would have given.

56. Another phase of the matter was shown by tests on boiler No. 18, when one of the baffles was in the condition shown in Figs. 112 and 113. Tests were made while in this condition and after repairs. Corresponding tests were also made at the throttle valve of engine *K*. The results were as follows:

Condition of baffle.	Range of moisture per cent. Different conditions of operation.	
	Boiler No. 18.	Engine K.
Imperfect	3.84 to 23.53	5.80 to 12.18
Perfect	1.66 to 5.80	2.25 to 2.99

57. The sample at the engine was taken at point *C* of Fig. 110; the sampling pipe did not extend into the main, the sample being gathered from the upper surface. From these samples it is not to be inferred that moisture entering engine *K* was as low as 2.25 per cent. But these records compared with those from boiler No. 22, would indicate that if there was moisture enough present, it would enter the calorimeter even with the flow of steam at high velocity.

58. Samples from the center of the steam pipe from boiler No. 14, taken under different conditions ranged from 1.66 to 3.04 per cent. moisture. This and No. 16 are the ones most affected by the drop in pressure.

59. Fig. 108 shows the steam header of group No. 3 serving engines *K* and *L*, with its various connections; 1, 2 and 3 are drip chambers, the location of which has an important influence on the amount of water going to the engines. With engine *K* out of service and *L* in operation, more water goes to it than when both are running, but with both in operation, engine *K* always received

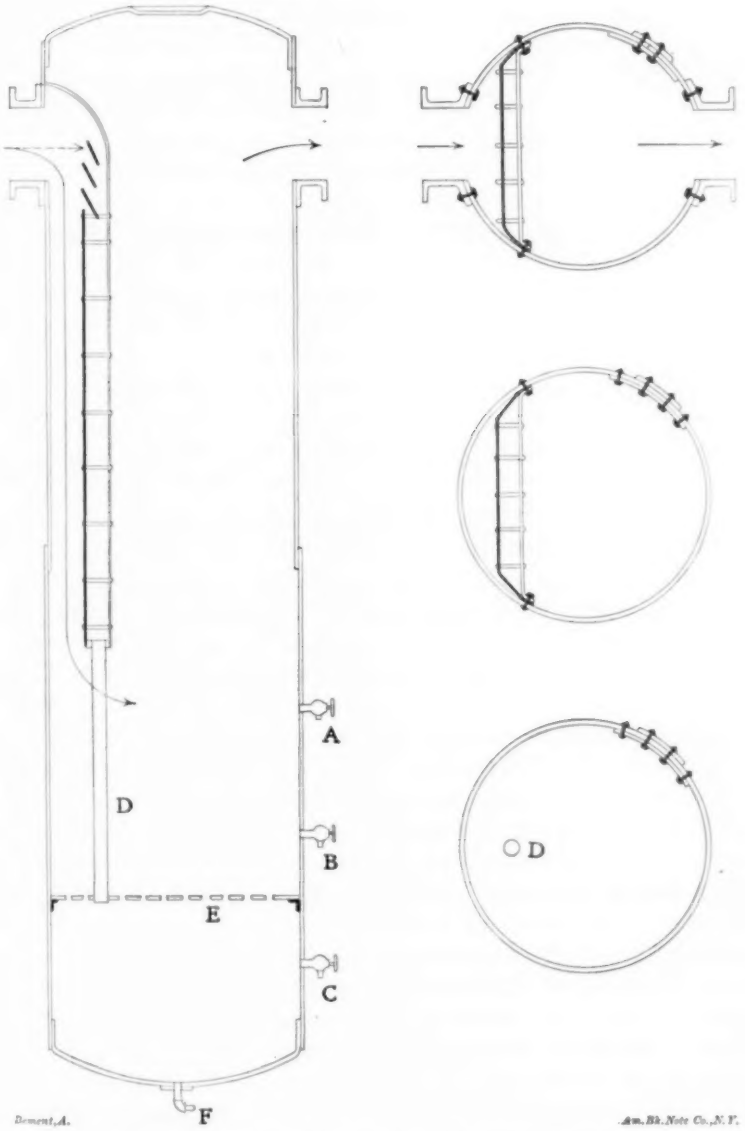


FIG. 117.—LARGE SEPARATORS. ENGINES K AND L.

more water than *L*. The explanation is found in the location of the drip chambers, No. 1 being so placed that the water from boilers Nos. 21 to 24 cannot reach it, neither can it flow to cham-

ber No. 2 in any great measure, because at valve opening the flow converges from both directions to the connection to the engine. But when both engines are in operation K induces a flow which removed some of the water from the vicinity of the outlet to L. With engine K the larger portion of the water from boilers Nos. 17 to 20 flows to it because there is no intervening drip chamber, but that portion going in the direction of engine L has an opportunity to enter chamber No. 2. It is apparent that the location of these drips is faulty, that No. 1 should have been where 4 is indicated, and that there should have been one as shown at 5.

Performance of Large Separators.

60. The above experiments and tests were made when the small separators were in use. Since that time very much larger ones, measuring 4 feet diameter by 17 feet long, of a unique design have replaced them, which are shown in section by Fig. 117. The upper portion being divided into two parts by a hollow partition, making the area of the inlet side very small, while the outlet side is very much larger. As shown there is an opening at the top of the partition, fitted with metal plates at an angle. It would appear that the operation of this device is based on the assumption that the water would leave the rapidly moving steam and flow directly into the top of the partition, from which it would pass by way of the 2-inch pipe D to the chamber at the bottom, below the perforated partition E. For this to occur, however, would require that separation take place in a much smaller space than was afforded by the small separators, which were themselves too small, therefore it necessarily follows that the entering water must continue with the steam through the inlet passage, into the large volume of the outlet side, where separation would occur if reduction of velocity be sufficient. To determine whether separation was effected, gauge cocks A and B from the locality of assumed dry steam was opened, with the result that the escape of water and steam alternated each other. But from cock C leading from the water chamber dry steam issued continually. It is probable that if these separators had been so connected that the steam enter what is now the outlet, the result would have been better, because if the velocity in the large chamber should be low enough to allow of separation it would be more readily attained in a downward flow.

61. The indications to the present time are that all of the water entering these separators leaves again with the steam. But owing

to their large volume effective steam storage space is very much increased, and for this reason less water leaves the boilers, their installation being a partial remedy only.

62. For water to separate from moving steam requires that the velocity be sufficiently low for it to occur, and if the quantity of steam be great, it necessitates an inconveniently large space. Therefore it is better to accomplish separation at the boilers, where in this case it could be divided up among the 24 boiler drums with the added advantage that the water does not require to be returned to them. This the high form of baffle with deflectors would accomplish in a most satisfactory manner.

DISCUSSION.

Mr. H. H. Suplee.—In regard to the matter of the behavior of steam separators, it is important that they should be properly proportioned to their work. In some investigations recently made in England it has been clearly shown that the effectiveness of the separation depends to a great extent upon the velocity of the passage of the steam through the apparatus, and that for every separator there is what may be called a critical velocity, above which the separation is not effected. This critical velocity varies for separators of different designs, but it exists in all types, and should be taken into account in selecting a separator for any given service. It might be well to have this critical velocity determined for every size of separator and so marked on it that the piping and connections could be arranged accordingly.

Mr. R. H. Rice.—I would like to ask if any theory has been formed to account for the fact of the critical speed mentioned by Mr. Suplee. The theory on which certain separators have been designed is that in these separators the direction of the steam is suddenly changed and the water, which is heavier than the steam, is thrown out during this change of direction. If a critical speed occurs above which separation does not take place, the theory on which these separators have been designed is manifestly erroneous. Undoubtedly at sufficiently high speeds the water contained in the steam becomes reduced to the form of very finely divided spray, which it would be very difficult to separate by centrifugal means.

Mr. A. A. Cary.—A number of years ago, when connected with the Root Boiler Co., my attention was directed to the delivery of what I considered too large a percentage of moisture in the steam

from the several steam outlets located at the rear of the steam and water drums of this boiler.

You will probably recall the construction of the Root boiler at that time, in which the bank of inclined water tubes delivered their steam and water into the front heads of a number of steam and water drums 14½ inches in diameter. This steam and water was delivered through curved pipes (or U bends). The 14½-inch drums previous to my investigation had been placed perfectly level.

From the upper and rear ends of these several 14½-inch drums the steam escaped into a large cross drum placed above the steam and water drums.

This cross drum was supposed to contain nothing but a supply of steam, but with the delivery of water from the 14½-inch steam and water drums, the cross drum acted more or less successfully as a separator; the water precipitated in this cross drum being carried off by a system of drainage pipes.

With such undesirable conditions existing, I was led to conduct a series of tests on somewhat similar lines to those described by Mr. Bement.

I began these experiments with glass models, which clearly pointed to the source of the trouble, but in such an energetic way that I wondered how the boiler could possibly deliver dry steam as I knew it actually did.

To obtain further information, I next placed a number of gauge glasses along the length of the 14½-inch steam and water drums as I have shown in Fig. 118, connecting these gauge glass fittings into the top and bottom of these drums.

I then filled the boiler so as to bring the water level to the center of the drums and tied black cords around the gauge glasses at the true level found by the surface of the water at rest. Afterward, when the boiler was in operation, I measured from these cords to the various water levels found in each of the gauge glasses and so was able to compare such levels with those obtained in my glass models, which models, I found, exaggerated the trouble very materially.

When the boiler was in operation, the water line in the steam and water drums ceased to be level, and from the indication of the water gauges I found it assumed an irregular line similar to that shown in the upper dotted line in Fig. 118.

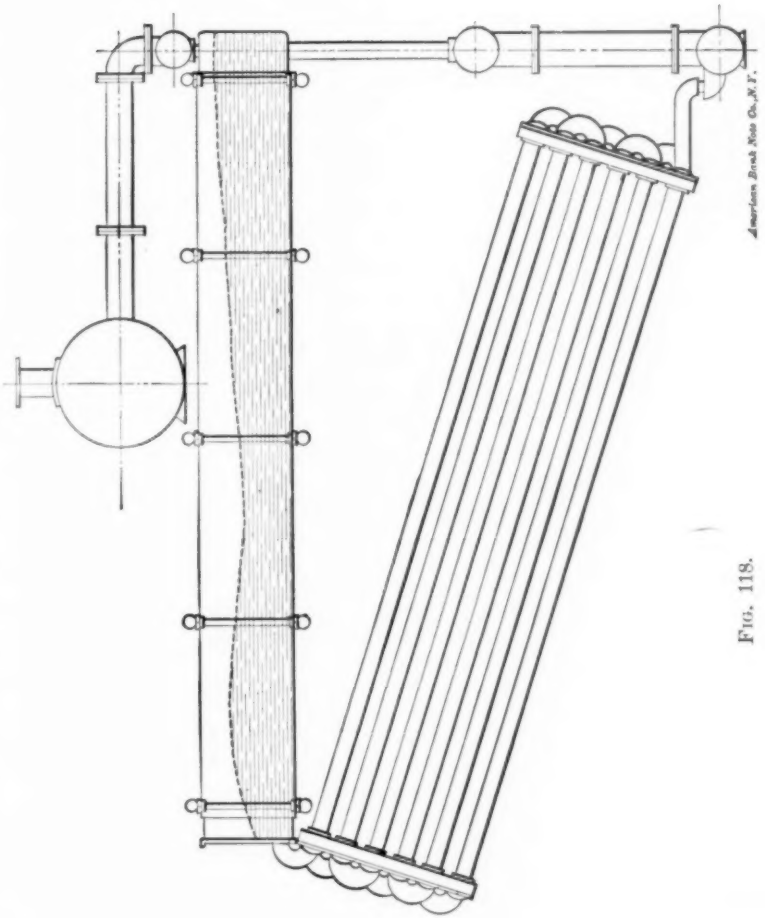
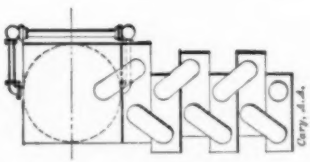


FIG. 118.



The steam and water from the inclined tubes was delivered with such velocity into the front end of the steam and water drums that it caused the water level to be considerably elevated at the rear end, and with a pulsating movement it would occasionally throw water upward and out of the steam outlet.

This trouble I succeeded in overcoming by elevating the rear end of the 14½-inch drums until the water at that position with the boiler in operation seldom was raised above the center of the drum. When in this elevated position, I never found the raised rear end of the drum dangerously unprotected by water.

This change did not, however, overcome all my troubles, as I still found water occasionally banking up at the rear end of the drums, and my glass model showed the cause very clearly.

Directly beneath the top steam outlet of the drum, a water circulating pipe was placed, down which passed the water discharged into the steam and water drums (from the bank of inclined tubes). As long as this water discharged freely down these "down takes," no trouble occurred with the improved setting, but I found that occasionally a vortex action occurred in the flowing water, directly over the "down take," similar to the action one often sees after the stopper is withdrawn from the bottom of a filled wash basin.

With such swirling motion, no water is discharged down the center of the escape pipe, but all escaping water hugs the circumference of the discharge opening, with the hole in its center materially diminishing the discharge capacity of the outlet.

I succeeded in remedying this trouble by inserting an open wire cage into the top of the down-take pipe, which cage was similar to those used in house gutters to prevent floating matter from being washed down and stopping the leader pipe.

This cage effectively broke up the vortex action of the water, and an ample discharge followed.

After these two changes we had no further priming trouble with the boiler.

Glass models may be very useful in investigating boiler troubles, but my experience has taught me that they are very apt to magnify such troubles, especially if we are seeking the cause of priming and so, if priming can be prevented in the model, it is pretty surely eliminated in the large boiler itself when similar remedies are applied there, although such remedies may in actual practice be more elaborate than actually required.

Glass models used to show circulation in boilers are often quite wrong in the results they are intended to exhibit, for the same reason that the troubles described above are exaggerated in such boilers.

The small tubes and low pressures used in model boilers prevent the circulation of well-mixed bodies of steam and water; such as may be compared with soda water, and when considerable heat is applied we find, instead of such a mixture, separated slugs of steam and water passing up the tubes.

Of course the circulation of water in a water-tube boiler is due to the velocity head obtained by the greater weight of the column of water descending along the rear of the boiler as compared with the lesser weight of the column of water ascending along the front of the boiler.

If the ascending leg of circulating water and steam be kept an unbroken column of water (irrespective of the bubbles of steam it contains), the circulation will be much slower than if the ascending current be made up of alternating slugs of steam and water.

The former condition is the one commonly found in actual boilers while the latter condition is the one found in glass models.

Most glass models are operated under atmospheric pressure, while the boiler itself may be operated under 200 pounds gauge pressure, in which, of course, the size of the separate steam bubbles are very much smaller.

I can hardly agree with Mr. Bement's criticism of steam calorimeters, providing they are properly applied. More errors are obtained in steam calorimeter work by improperly sampling the steam than from any other cause.

If Mr. Bement bases his criticism upon the results obtained in his glass models, as compared with the results obtained at the boiler itself, of course a considerable difference will be found, but as I have already pointed out, this difference is due to the exaggerated results obtained in the small model.

Professor Kent.—I have not been able at a glance at the paper to tell which place he recommends for placing the baffles.

Mr. John J. Hoppes.—I wish to say that I attribute the trouble in this case and also the difficulty of taking samples for calorimeter determination to a fact mentioned in the paper, and that is the following of the water along the surface of the pipe. This also occurs in boilers, separators and reservoirs.

To illustrate this in a simple manner, I will mention a case I

had a few years ago in which a return tubular boiler with a very high water line was giving trouble. The owner stated he had the usual so-called dry pipe inside the boiler.

I had a little device made and placed in the boiler, shown in Fig. 119. It consists of a bell-shaped nozzle extending downward and having its outer edge rolled up to form a trough as

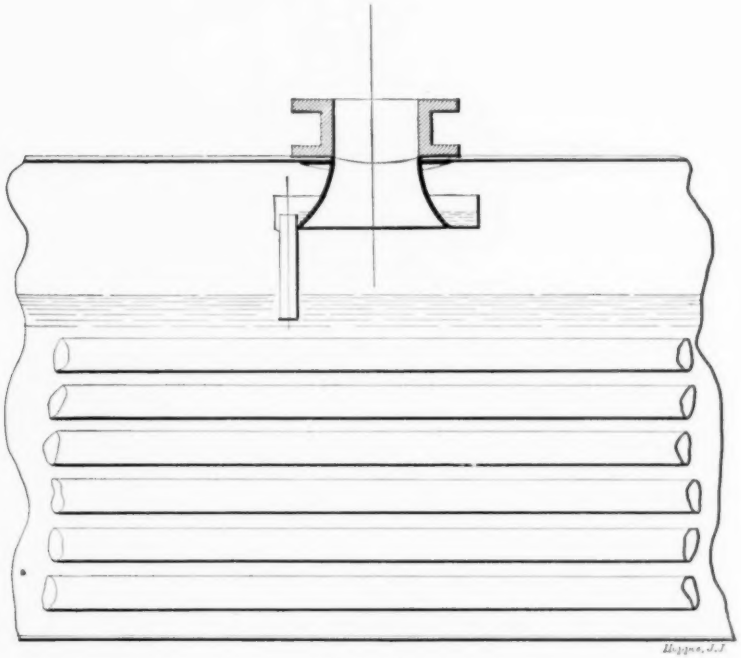


FIG. 119.

shown. A small pipe was tapped in the bottom of the trough, the upper end extending above the bottom of the trough about 1½ inches, so as to retain a small amount of water, and the lower end of the pipe was carried down below the water line.

The water in the trough intercepted the water flowing on the surface toward the steam outlet and the small pipe carried the surplus to the water space below.

This remedied the trouble, and later on in designing separators I followed out this same principle and found it very effective.

The large separator shown in Fig. 117 could have been made effective had it been made differently in reference to the intro-

duction of the steam. Instead of the diaphragm being carried down near the shell on the inlet side, it should have been gradually carried out to the center of the shell, and a trough placed at the bottom with a pipe to carry off the water to prevent it spraying off into the steam space.

It would also be well to protect the outlet with an intercepting trough.

The modified form is shown in Fig. 120. Many separators with small working parts only tend to break up the water and spray it off in the steam, and calorimeter tests, where the sample is taken in the usual way, often show higher saturation after passing than before entering the separator.

Now when you carefully intercept the water flowing along the surface of the pipe and carry it to the bottom instead of spraying it into the separator, much better results are obtained.

Mr. Cary.—I have had some experience with water creeping upwards around the shell of a boiler and causing wet steam.

Among the first of the Root boilers built with large steam and water drums we were obliged to omit the customary cross steam drum and, with a view of obtaining dry steam, we placed a "dry pipe" longitudinally along the top of the steam and water drum.

This "dry pipe" was about 3 inches in diameter and had a series of slits cut in its top side, while its solid under side was presented towards the water (carried in the lower part of the drum).

A complaint concerning moisture in steam was received from the purchaser of this boiler and, on investigation, I could not account for this trouble by any other theory than that of water creeping upward along the shell, so, as an experiment, I had the slits in this dry pipe turned downward (and looking towards the surface of the water) and the trouble complained of ceased.

Mr. George M. Brill.—This discussion having drifted to the subject of steam separators, I am led to refer to some tests and experiments made several years ago on practically all the separators on the market. These tests covered a range of qualities from dry steam to 40 per cent. of water. Of course it was found that the principle of separation depends upon change in direction of the steam, increased volume producing decreased velocity and centrifugal force. Each separator has an equation based upon the manner these principles are used and combined. In several cases it was found that, while the separators might separate for

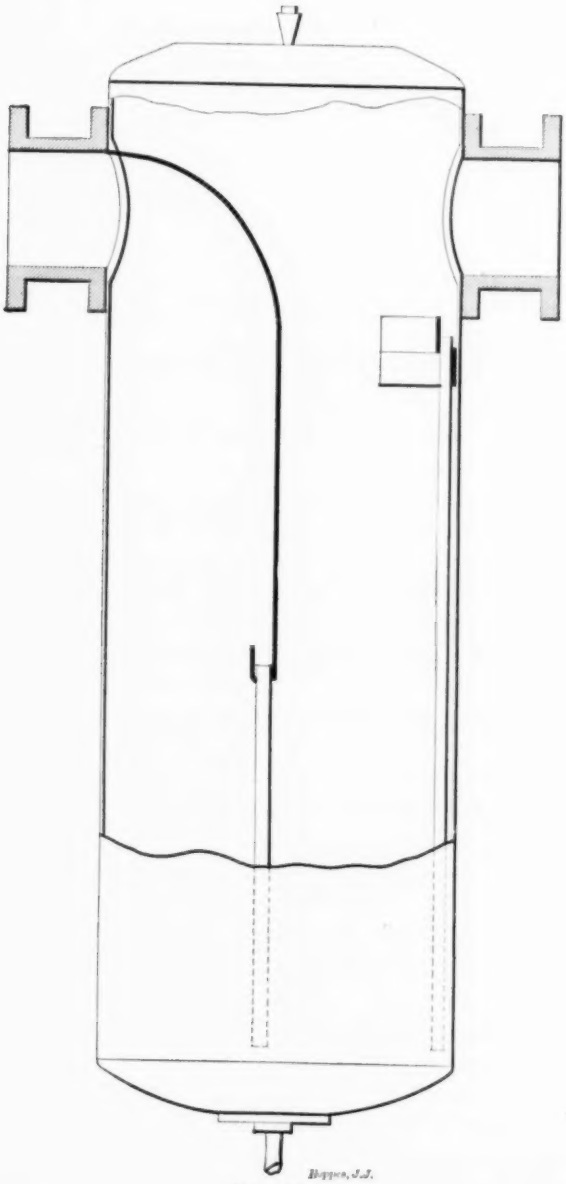


FIG. 120.

the instant, the water was again picked up and carried along with the steam because no provisions were made to keep it away from the steam after separation, and it is my opinion that many of the separators now on the market are inefficient and ineffective in performing their function because little or no attention has been given this feature. The water is taken out of the steam but not kept out. Some of the separators on the market would seem to indicate that their design is largely the result of the theory of permutations and combinations. In my opinion the separator is a more important element in steam plants than many engineers and users appreciate. It frequently has an opportunity to prevent a bad and over-worked boiler from wrecking a good engine and more frequently to improve engine economies by keeping water out of the cylinders.

Mr. John J. Hoppes.—I would like to call attention to the fact that a boiler often shows a false water line, owing to the manner of connecting the water column. I had a case of a separator installation where a vast amount of water came over with the steam, and I found that the water connections to the water columns on the vertical boilers from which steam was taken was connected near the bottom of the boilers. The water in the pipe being almost cold counterbalanced a much higher column in the boiler, and on blowing off the water in the pipe and showing the true level, a difference of about 11 inches was observed, so that the water in the boiler was just that much too high.

I had the same experience with several of the earlier Root boilers, the method of connecting the water column showing about 7 inches less than in the drums.

I called the manufacturer's attention to this and it was corrected. I think over twenty cases of this kind have come under my observation.

Mr. C. B. Rearick.—In line with the question of carrying water out of the boiler, I made a test in a little model in which I had glass heads arranged in the drum, which was of four-inch pipe, as a matter of fact, and there were four 3-inch tubes, nine inches long. These four tubes were connected to a header. I had six 1-inch wick alcohol burners, which supplied a very high heat, evaporating at the rate of 9 to 10 pounds of steam per square foot of heating surface.

One point brought out by Mr. Bement's paper is the fact that a sudden demand for steam on the boiler takes part of the water

with it. I think that is one of the worst things to contend with in steam generators. The experiment I made seemed to prove it very effectively, and while I experimented at atmospheric pressure, exaggerating the case to a considerable extent, it showed what actually happened. For instance, I allowed the pressure to build up by plugging the steam opening. In delivering the steam the water was piled up at the steam outlet and thrown against the top of the drum. When the steam was generated with the steam outlet open, the steam was practically dry. By stopping up the steam outlet the pressure was allowed to build up to from 5 to 10 pounds. The plug was then pulled and the whole mass of water had a tendency to flash. This caused a very furious delivery of steam and water from the tubes and header which was so intense that water was carried out with the steam to such an extent that some was carried to a ceiling 8 or 10 feet above, which shows the enormous effect of the sudden demand for steam or lowering of the pressure suddenly.

In the diagram on page 339 I rather question Mr. Bement's point that the generation takes place at the rate of 2,400 horse power with reference to the capacity it starts with. From the fact that generation must continue in order to build up the pressure in the steam drum, steam mains, etc., even when there is no real demand for steam, it would seem to follow that this difference could not be so great. For instance, we make a demand for steam and it is cut off; the pressure has been reduced and you therefore have a reservoir of steam which must be raised in pressure at the same time you are putting heat in the water. I believe there is a marked result, as Mr. Bement states, but I don't believe it is as extreme as he shows in the diagram.

*Mr. A. Bement.**—The remarks by Mr. Suplee interested me very much, and his definition of the condition at which separation ceases as the critical velocity puts in form some more or less vague ideas which I have had.

In connection with Mr. Rice's remarks, I suppose we may observe some of these effects in wind storms when air at high velocity carries large amounts of dust or sand. One of the most effective means of removing dust from blast furnace gas is to pass it through a large chamber where the dust will drop out of suspension if the velocity is sufficiently reduced. Such a scheme

* Author's Closure, under the Rules.

has been employed with success, I believe, in a case where much dust was discharged owing to high velocity from chimneys.

I have observed the tendency for the water to pile up at the back end of the drum, as Mr. Cary has shown in the sketch, and believe it to be greater in some than with this illustration of the Root boiler. The vortex action at the down-coming pipes at the rear of such boilers I think may be a matter of very great importance, and am inclined to question if this may not be an explanation of some of the cases of burned tubes of the lower row.

I assured Mr. Cary that I made no criticism of calorimeters, but I did make some statements about unreliability of methods of sampling, and my conclusions are, that if a boiler does discharge water with the steam, the only way to ascertain the quantity is to pass all of the steam through a separator of such large capacity that the flow will be below its critical velocity mentioned by Mr. Suplee, and to make correction for radiation. Mr. Cary, I take it, does not think that the calorimeter was properly applied, and this brings up the question what is proper application? I believe the orthodox method is to take the steam out of the center of a vertical pipe. Now, this might be called a correct method or not, depending upon whether the boiler-makers or the purchasers' interests were to be served. If the sample is taken from the surface of the pipe as well as the interior, and it shows above one or two per cent. moisture, are we to say that it is from radiation, and if so, where are we to get enough radiation from a few feet of covered 8-inch pipe to amount to a noticeable percentage of the great amount of steam flowing through the pipe? From my standpoint, the sampling was correct. It was my purpose to find water, if possible, and I was bound by no tradition as to method, but I had much more accurate means than that afforded by the calorimeter, which were the engines themselves. I could not make myself believe that, when the sample showed about 1.5 per cent. moisture, the actual presence of water in the engines was imaginary; in fact, I did not try to; neither did I believe that, when the sample showed 17 per cent. moisture with a boiler at low capacity and a small load on the engines, such sample gave the correct percentage of moisture in the steam leaving such boiler. In these investigations, I did everything practicable to study the trouble, but with all the evidence afforded by the engines, calorimeter and model boiler I arrived at no conclusions as to percentage of moisture. I knew before the experiments began

that it was sufficient to make trouble; in fact, it required no experiments to illustrate this. I did not come to any conclusion about percentage of water discharged from the large boilers from evidence furnished by the model, but this boiler, with the glass drum, showed very plainly how the water did leave, and how it could be prevented, and the violent ebullition in this model was useful as illustrating the great value of the improved form of baffle which was devised.

In answer to Professor Kent's inquiry, the best position for the baffle is shown in Fig. 95, as designated High Baffle; also in Figs. 102 to 105.

Experiments were made with my model, similar to those mentioned by Mr. Rearick. The steam outlet was provided with a three-way cock leading to the atmosphere; also to a drum in which a vacuum existed. When steam began to discharge to the atmosphere after the blast flames were lighted, the gas was shut off, leaving the water at a temperature due to atmospheric pressure, then the cock was turned to the vacuum chamber and the water burst into violent ebullition for a very brief instant, until the pressure equalized. These experiments were made with different drops in pressure, and large and increasing amounts of water were carried out as the drop in pressure became greater. The experiments were very convincing and proved what we knew to be true.

As to the correctness of my diagram illustrating the rate of evaporation caused by drop in pressure, Mr. Rearick can, of course, calculate it for himself. What I have shown, however, illustrates the matter as I wished to present it.

No. 1070.*

A TWIST DRILL DYNAMOMETER.

BY WILLIAM W. BIRD AND HOWARD P. FAIRFIELD, WORCESTER, MASS.
(Members of the Society.)

1. The introduction of the high-speed twist drill into modern machine shop practice has brought with it many new problems, and to supply some of the data necessary for their solution the machine which is the subject of this paper was designed and constructed. It was taken as a thesis subject by Mr. George F. Read, Jr., of the class of 1904 at the Worcester Polytechnic Institute, and some of the preliminary work was done by him. All the experiments here reported, however, were conducted by the authors at the Washburn Shops of the Institute.

2. The machine is really a double dynamometer, measuring the twist or moment and the thrust of the drill. Two views are given of the machine in Figs. 121 and 122, which show the actual arrangements of the parts during the tests. It consists of a chuck with trunnions mounted on roller bearings, so that it is free to revolve about its axis at the same time it is free to move in the direction of its axis, being mounted on a carriage which is on rolls resting on the frame of the machine. This frame was placed on the table of a milling machine so that the axis coincided with the center line of the drill, which was driven by the spindle of the milling machine. The thrust of the drill is transmitted through the trunnion of the chuck to a plunger which is fitted into a long cylinder filled with oil. At the end of this cylinder is a smaller one also carrying a plunger. This small plunger is attached to a ball-bearing piston in a regular indicator. The drum of the indicator moves proportionately to the feed so that the card taken is a record of the thrust during the drilling of a hole. The chuck is held

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

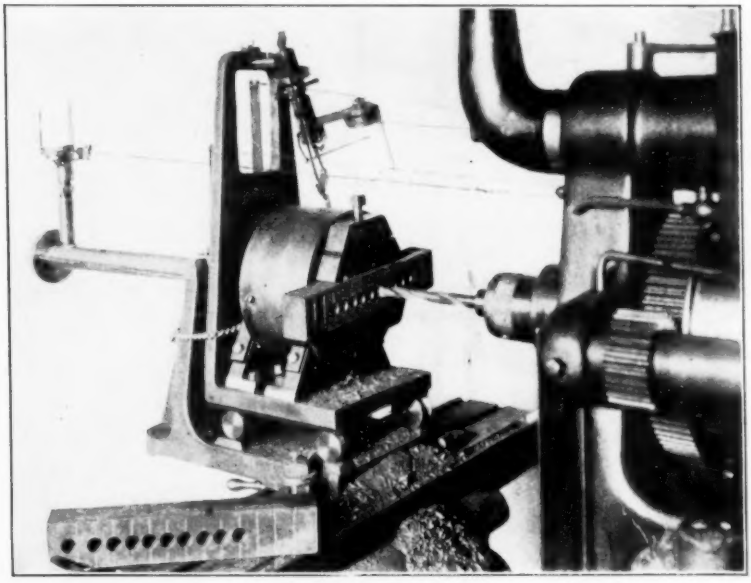


FIG 121.

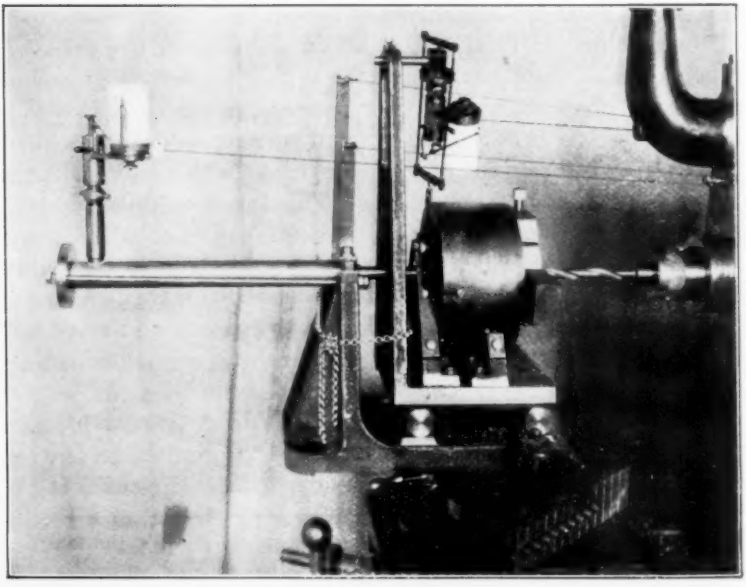


FIG 122.

from turning by a steel band which is attached by a yoke to a second indicator, and a card from this one shows the twisting moment at each stage of the operation.

3. Both indicators were calibrated before and after the tests.

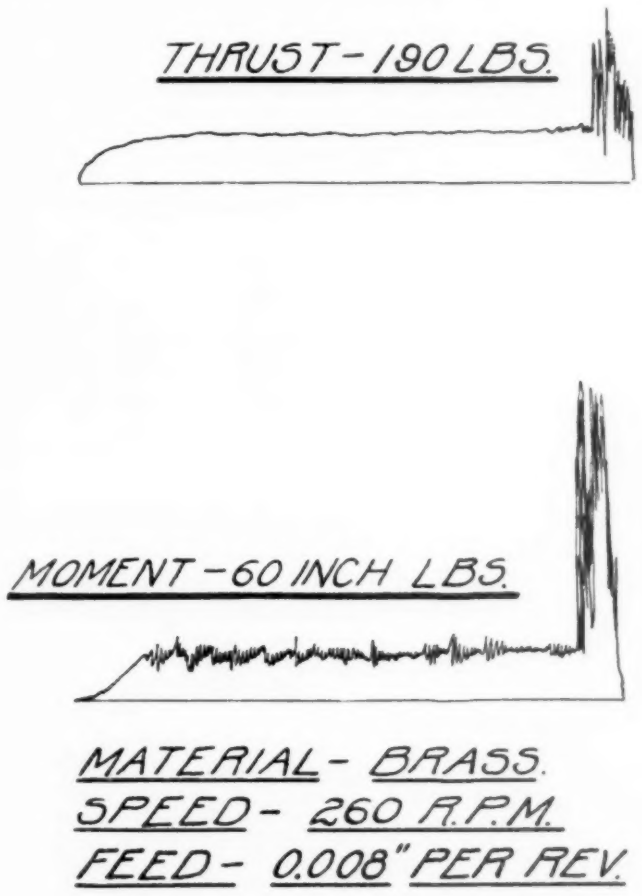


FIG. 123.

The correct moments were obtained by hanging known weights over the chuck and noting the rise and fall of the pencil point. The thrust cylinder was placed in an Emery testing machine and known loads applied and a corresponding record taken on the cards. From these tests, curves were plotted so that readings from the cards could be readily reduced to pounds.

4. The following cards, Figs. 123, 124, 125 and 126, are samples of the cards taken, and show a comparison of drilling in different materials as noted. In each case the feed was 0.008 inches

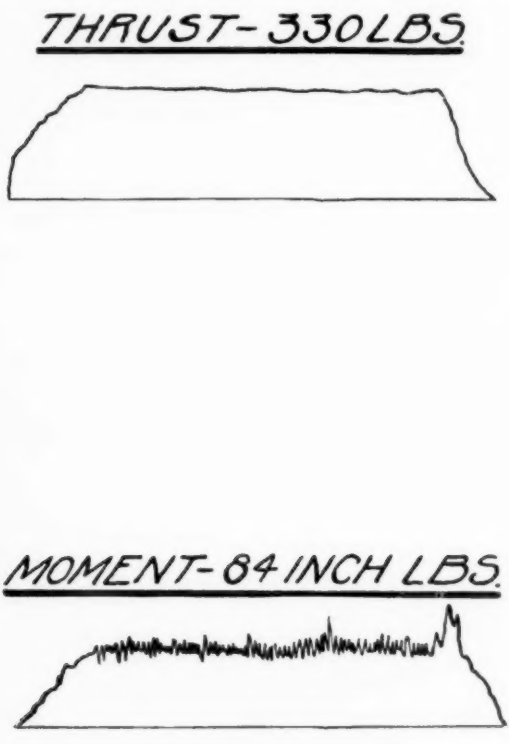


FIG. 124.

per revolution, the number of revolutions 260 per minute, and the size of the drill $\frac{3}{4}$ inch, with standard angle of 59 degrees.

5. It was decided to take up first the drilling of cast iron with high-speed steel. Blocks of soft gray iron were obtained and tested in the machine itself so as to get a number for each set of experiments as near the same degree of hardness as possible. A

$\frac{5}{8}$-inch Novo steel drill was taken as a good representative of the new steels.

6. The first experiments were on the effect of speed or number

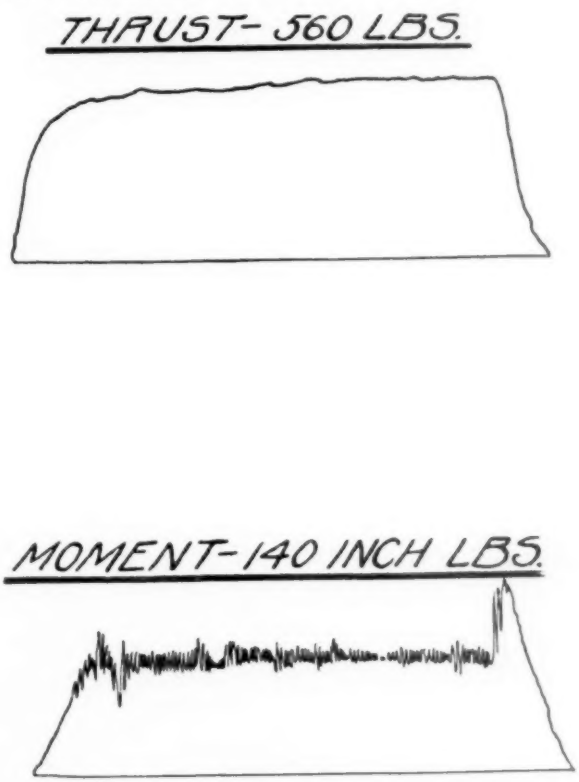


FIG. 125.

of revolutions per minute, all other conditions remaining the same. The revolutions were varied, from 140 to 600, but no material difference was shown by the cards in either thrust or twist. In other words, the power required to turn the drill varies directly with the number of revolutions, while the thrust does not increase

with the speed, but depends upon the feed or the advance per revolution.

7. The next set of experiments was made to determine the relation between thrust and feed, the revolutions per minute remaining constant. From the first set of experiments it was shown that the limit of speed would depend upon the endurance of the drill,

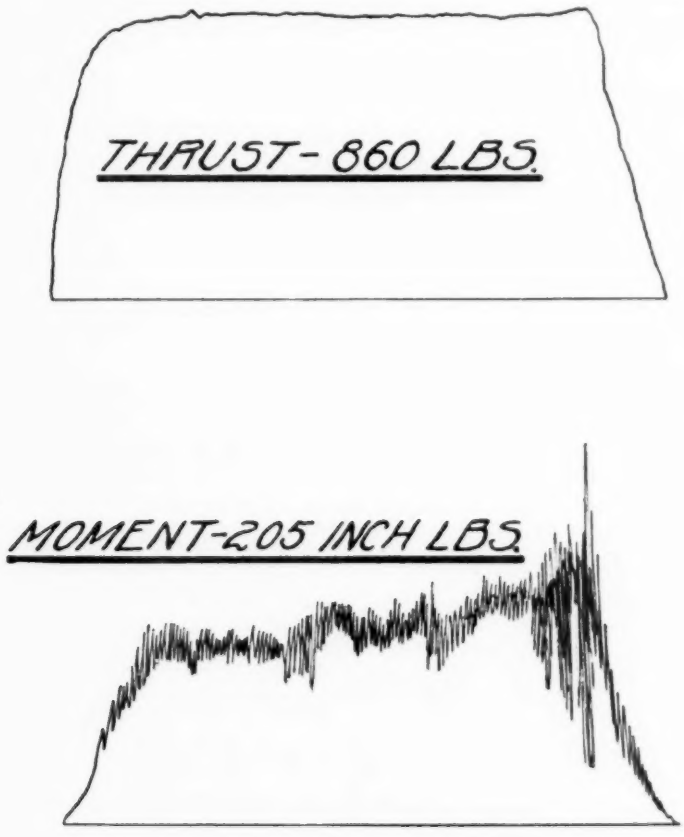


FIG. 126.

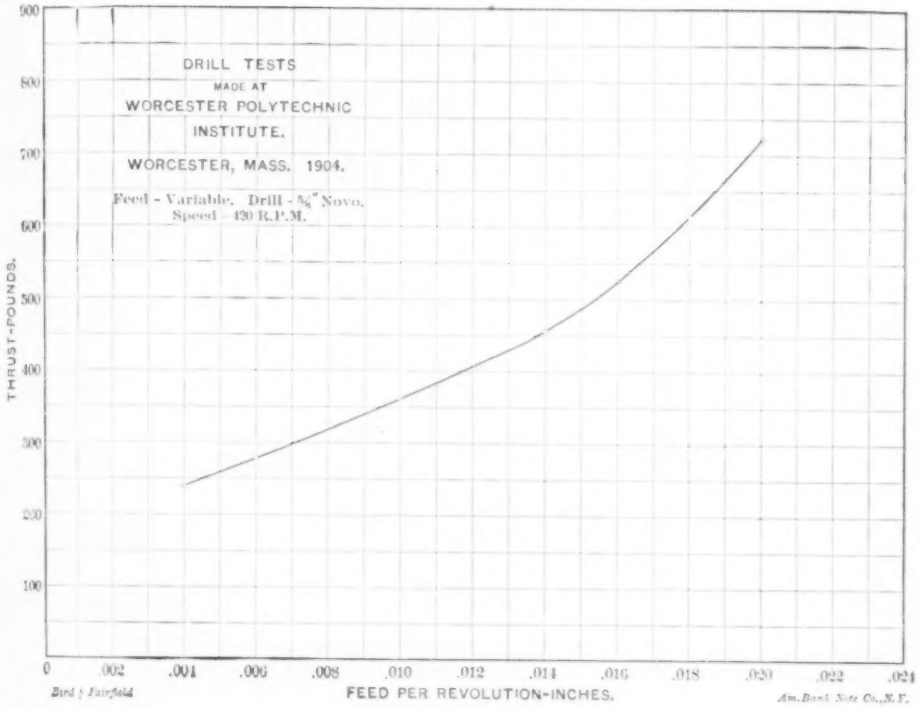


FIG. 127.

and with heavy feeds 420 revolutions was not far from this limit. Accordingly the second set was run at this speed, which for a ⅝ inch drill is about 70 feet per minute for the cutting rate for the outer edges and for 0.020-inch feed a rate of drilling of about 8 inches per minute.

8. The range of feed was taken from 0.004 to 0.020 inches per revolution, the drill at the coarsest feed being somewhere near its limit. These feeds were all positive, a train of gears being substituted for the regular belt drive. The following table shows the results:

TABLE I.

Feed:—inches per revolution—								
.004	.006	.008	.010	.012	.014	.016	.018	.020
Thrust:—total in pounds—								
240	280	330	360	410	460	525	620	725
Moment:—total in inch pounds—								
55	67	83	92	103	115	124	132	138

9. Fig. 127 gives the same data for thrust and feed in the form of a curve and shows that the thrust increases very rapidly with the coarser feeds.

10. Fig. 128 is a curve giving the relation between moment and feed, and shows that the moment does not increase at the same rate as the thrust. This would seem to indicate that less power

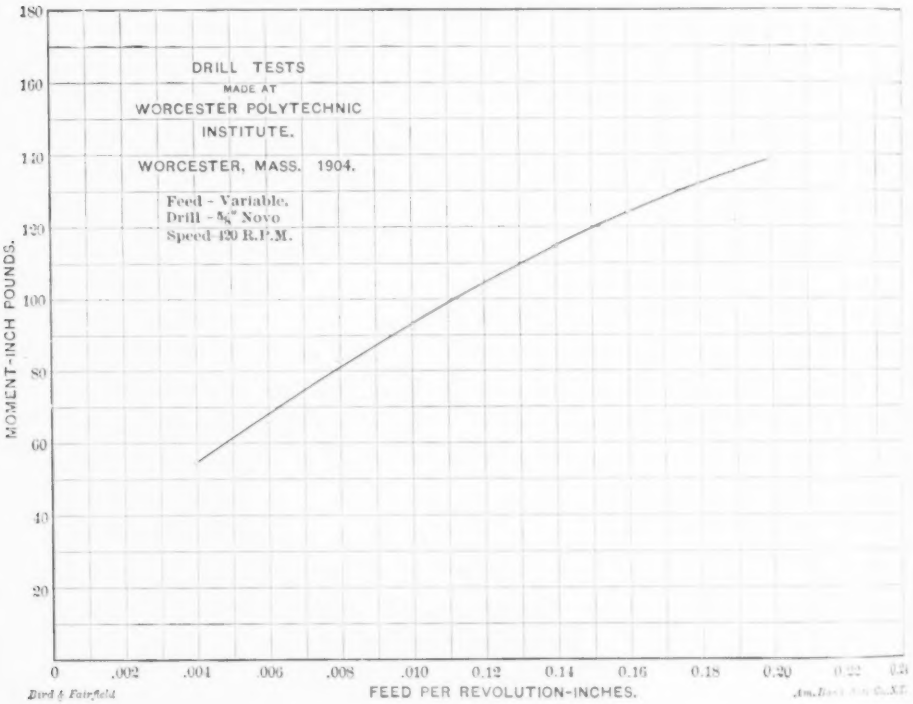


FIG. 128.

is required to drill a given hole in a given time by increasing the feed per revolution than by increasing the revolutions. For example, the drilling of a hole in a 1-inch plate in 10 seconds could be done by running the drill 600 revolutions per minute and feeding 0.010 inch per revolution, but would require more power than by running at 300 revolutions with a feed of 0.020 inch.

11. The question of the proper angle of the drill was next considered, and a set of experiments made with a constant speed and feed, the half-angle varying from 37 degrees to 70 degrees, 2

degrees less and 11 degrees more than the standard 59 degrees. The following table shows the results:

TABLE II.

Angle of drill.....	37°	45°	50°	55°	59°	65°	70°
Thrust in pounds.....	215	225	240	260	290	325	375

12. Fig. 129 is a curve for the same and shows that the thrust would be decreased by having more of a point on the drill. With

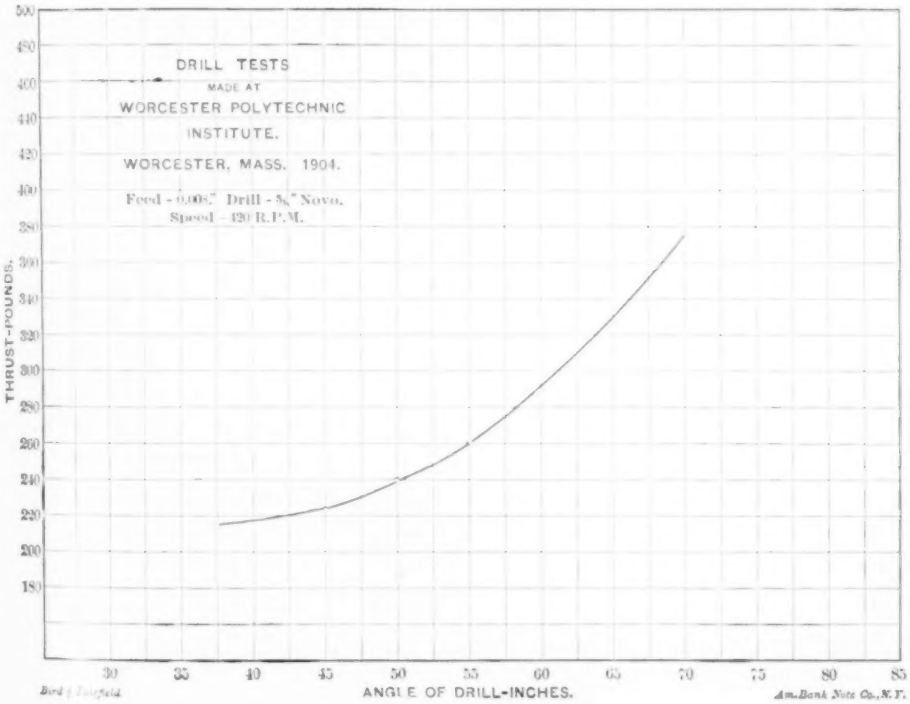


FIG. 129.

an angle of 37 degrees, however, the drill would not stand up on repeated work. At 45 degrees it seems to do the work as well as at 59 degrees and with much less thrust. This would suggest a change in the standard angle for the new steels. The moment for the various angles remained practically constant, so that the driving power does not change with the angles of the drill.

13. Another interesting point in drilling which can be shown

with this machine is the effect of first removing the center of the hole by the use of a small drill. Ten holes were drilled in a bar of cast iron with small drills, ranging from No. 53 to $\frac{1}{2}$ inch in diameter and then each one counterbored with a $\frac{5}{8}$-inch drill at 420 revolutions per minute and a feed of 0.008 inch per revolution.

Table 3 gives the thrust in each case and shows that a hole 1-10 inch in diameter takes off about one-half the thrust for a $\frac{5}{8}$-inch drill.

TABLE III.

Size of first hole—										
.000	.0595	.0760	.0935	.120	.157	.189	.221	.250	.375	.5
Thrust for counterbore—										
340	190	180	170	155	145	130	120	90	70	30

14. The range of experiments is almost limitless, and to cover the entire field for feed, speed, angle of drill, etc., for various materials and all kinds and sizes of drills will require much time and many tests. Some of the results so far acquired are here reported with the hope that others may add to them, or suggest lines for further experiments.

DISCUSSION.

Mr. Harrington Emerson.—Even in academic experiments nothing adds so much to their practical value as tables and data which, however approximate, can be immediately used, until something better is furnished, by those in practical life who have no time either to make experiments or deduce unexpressed conclusions from data furnished by others.

For the everyday man, three times the radius is generally close enough for the circumference of a circle, while 7 to 22 answers for ninety-nine cases in a hundred, and 113 to 355 for all cases except one in a hundred thousand. For the final case, we need the exceedingly refined instruments made by our president, Mr. Swasey.

As a practical man, I regret that Prof. Bird did not tabulate suitable and desirable angles, speeds and feeds, and also state what initial hole he recommends in order to accelerate work, lessen wear and tear on drills and economize power.

Mr. Fred. W. Taylor.—I should like to ask Prof. Bird a question. What does he mean by the words "stand up," as used by him in paragraph 12?

Prof. Bird.—I would say in regard to this that 37 degrees of course gives a somewhat pointed drill, and what we mean is that this point breaks down after drilling two or three holes.

Mr. Fred. W. Taylor.—In corroboration of what Prof. Bird says, I would state several years ago Wm. Sellers & Co. adopted 45 degrees as the best angle for drilling, having found that 45 degrees was much better than 59 degrees for everything except very hard work. I think every one who has experimented on the subject has concluded that on exceedingly hard steel it is necessary to use a blunt drill. The angle between the portion on which the chips lie that angle in turning and in drilling the machine generally.

Mr. S. W. Baldwin.—I call attention to the fact that in looking at this specimen of cast iron drilling one lip of the drill is about 1-64th inch longer than the other and that for making experiments for record those who make the experiments ought to use more care in their workmanship. It is very clearly seen here that in the drilled specimen the length of the lips of the drill are not alike.

Prof. Bird.—The sample Mr. Baldwin refers to was made with a ½-inch experimental drill. We brought the sample here to show how one lip will cut a little more than the other when the web of the drill is not central.

Mr. W. J. Kaup.—It seems to me that there is nothing to indicate what has been gained by the experiments made by means of the dynamometer.

They have stopped short of establishing what to the mechanical world is all important, and that is, is there an angle that is economic for all metals, and if so what is that angle? Also what is the best method of overcoming the crushing effect on the point of the drill, and what shape should it be to cut and not scrape?

I have made very many tests with specially designed drills covering many weeks, and have nothing worth recording, for the reason that, as soon as I changed the slightest condition I destroyed the value of my data. What should be done, and I hope Prof. Bird and Mr. Fairfield will carry along the experiments, is to use the commercial twist drill, and those tools that the market is selling as stock goods, and not confine the experiments to specially designed tools and steels.

*Prof. Bird.**—All of the experiments we have here reported

* Author's Closure, under the Rules.

were made with a regular $\frac{3}{4}$-inch Morse steel drill which was bought in the open market. We have attempted to show something of how the power and thrust vary with the feeds. Actual values, however, are only true for drilling cast iron of the same degree of hardness as that used for each set of our experiments. Definite conclusions are not in order unless all conditions are known, and all conditions are not the same for any two cases. We do suggest, however, that high-speed steel drills should be more pointed than the present standard, and that whatever tends to decrease the thrust will lessen the power required to drive the drill, relieve the drill-press and add to the life of the drill.

No. 1071.*

INFLUENCE OF THE CONNECTING ROD UPON ENGINE FORCES.

BY SANFORD A. MOSS, WEST LYNN, MASS.
(Member of the Society.)

1. In this paper will be deduced a method for simply and yet exactly taking account of the influence of the weight and inertia of the connecting-rod upon the forces transmitted by the ordinary slider crank-chain, such as is used in steam and gas engines.

2. It will be shown that if the connecting-rod be considered partly as a rotating part concentrated at the crank-pin, and partly as a reciprocating part concentrated at the cross head-pin or wrist-pin, the effect of its weight and inertia will be taken account of almost exactly.

The method of treating the influence of the connecting-rod by assuming that it is partly rotating and partly reciprocating is of course not new. However, a rigid investigation of the matter, and a mathematical determination of the fractions into which the rod weight must be divided, is not given in any of the usual treatises on the subject.† In fact, statements concerning the matter are made in many treatises, which will be shown to be incorrect.

Notation.

3. All dimensions are in feet, all weights in pounds and all forces in pound's-weight.

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† Shortly after the first draft of this paper was written an article on "Der Beschleunigungsdruck der Schubstange," by Prof. Dr. R. Mollier, appeared in the *Zeitschrift des Vereines Deutscher Ingenieure*, Vol. XLVII., No. 45, November 7, 1903. A portion of the present subject is treated by wholly different methods. The final result which Professor Mollier reaches is identical with the statement made in the first sentence of paragraph 22 of the present paper.

l = distance from center to center of connecting-rod.
r = radius of crank, or half the length of stroke.
$n = l/r$.
W = weight of connecting-rod.
a = distance from center of gravity of rod to wrist-pin.
b = " " " " " " " " crank-"
Then $a + b = l$.
k = radius of gyration of rod about axis through center of gravity.
K = radius of gyration of rod about axis through wrist-pin. Then $K^2 = a^2 + k^2$.
$x, y,$ = coördinates of center of gravity of rod, referred to engine axis, as shown in Fig. 130.
ω = constant angular velocity of crank, in radians per second.
θ = crank angle, measured from engine axis. Then $\theta = \omega t$.
φ = connecting-rod angle, measured from engine axis.
$X, Y,$ = resultants of all forces acting on the rod, in the direction of, and perpendicular to the engine axis.
M = sum of moments about the center of gravity, of all forces acting on the rod.
P = reciprocating force (supplied by a portion of the piston pressure) necessary for the existing accelerations of the rod.
G = normal force at wrist-pin, *i.e.*, "guide reaction," necessary for existing accelerations.
C = radial force at crank-pin necessary for existing accelerations. This is the force equal and opposite to what may be termed the "centrifugal force of the rod."

$$V = \frac{1 - 1/n^2}{\left(1 - \frac{\sin^2\theta}{n^2}\right)^2 \left(1 + \frac{\cos\theta}{n\sqrt{1-(\sin^2\theta)/n^2}}\right)}.$$

We have the usual relation $l \sin\varphi = r \sin\theta$, whence $\sin\varphi = (\sin\theta)/n, \cos\varphi = \sqrt{1-(\sin^2\theta)/n^2}$.

Distribution of Inertia Forces.

4. We take as the forces acting on the rod, P, G and C, as shown in Fig. 130. These are those forces which necessarily exist in order that the rod may have the accelerations which result when the

crank rotates with constant angular velocity ω. They may be called the "inertia forces." Of course there are other forces acting upon the rod due to the transmission of force from piston to crank-pin. These mutually balance each other and take no part in the acceleration, so that they need not be considered.

5. The force P is a reciprocating force, furnished by the piston pressure. It is that portion of the whole piston pressure which is not transmitted through the rod to the crank-pin but which is absorbed by the rod inertia.

In other words, there is a definite pressure at the wrist-pin

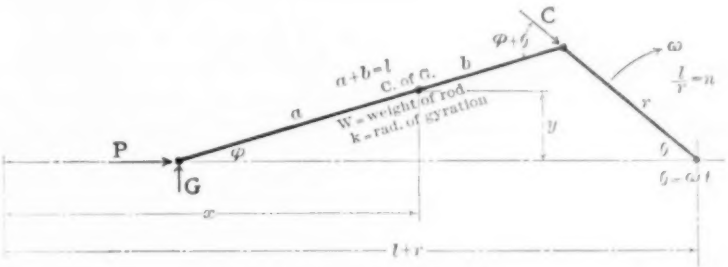


FIG. 130.

transmitted from the piston, whose values at each instant are given by the indicator-diagram, and a portion of which is intercepted by the rod, due to its inertia, forming our reciprocating force P. The balance of the piston pressure of the indicator-diagram, together with the accruing guide reaction, is exactly in equilibrium with the tangential pressure from the crank-pin, and the accruing radial reaction. This set of forces takes no part in the accelerations, and therefore, as already stated, need not be considered.

The rod acceleration also demands additional guide reaction, and radial force at the crank-pin, given by our forces G and C. C is an inward radial force upon the rod exerted by the crank-pin. It is also, in the opposite aspect, a so-called "centrifugal force" exerted outward upon the crank-pin by the rod.

6. The forces P, G and C are positive when in the directions shown in Fig. 130. These forces are of course variable functions of θ. At times they may happen to be essentially negative, and then will have directions opposite those given in Fig. 130. The

force P will turn out to be negative about half of the time. At such periods the rod inertia assists the indicated piston pressure in causing tangential pressure on the crank-pin, for the position of the mechanism shown in Fig. 130.

Fig. 130 shows the case for θ between 0 and 90 degrees. Our equations hold for all values of θ however, from 0 to 360 degrees. It is to be noted that no supposition is made as to whether the engine is horizontal, vertical, or inclined, and the equations hold for any case.

Method of Obtaining Expressions for the Inertia Forces.

7. In order to express the forces P, G and C in terms of θ, we first find the accelerations of the center of gravity of the rod in the direction of, and perpendicular to the engine axis, and also find the angular acceleration. We then apply the usual principle that the center of gravity moves as if all of the forces act upon it parallel to their actual directions, and that the rod rotates as if the center of gravity were fixed and all of the forces retained their original lines of action. This enables us to obtain from the accelerations the resultant forces X and Y in the direction of and perpendicular to the engine axis, and the force moment M about the center of gravity, all in terms of θ.

We may also express X, Y and M by resolving the forces P, G and C. By substituting their previously determined values, we then eliminate X, Y and M and express P, G and C directly in terms of θ. The reduction of these expressions gives our final solution.

Mathematical Evolution of the Inertia Forces.

8. We now proceed to a tedious mathematical treatment of the problem according to the method given above. We first obtain the acceleration of the center of gravity in the direction of the engine axis, *i.e.*, d^2x/dt^2.

We have evidently, from inspection of Fig. 130,

$$x = r + l - r\cos\theta - b\cos\varphi.$$

Differentiating,

$$\frac{dx}{d\theta} = r\sin\theta + b\frac{d\varphi}{d\theta}\sin\varphi$$

$$\frac{d^2x}{d\theta^2} = r\cos\theta + b\frac{d^2\varphi}{d\theta^2}\sin\varphi + b\left(\frac{d\varphi}{d\theta}\right)^2\cos\varphi.$$

Since $\frac{d\theta}{dt} = \omega$, this reduces to

$$\frac{d^2x}{dt^2} = \omega^2\left\{r\cos\theta + b\frac{d^2\varphi}{d\theta^2}\sin\varphi + b\left(\frac{d\varphi}{d\theta}\right)^2\cos\varphi\right\}.$$

By differentiating $\sin\varphi = (\sin\theta)/n$ we obtain,

$$\frac{d\varphi}{d\theta}\cos\varphi = \frac{\cos\theta}{n}, \text{ whence } \frac{d\varphi}{d\theta} = \frac{\cos\theta}{n\cos\varphi}.$$

Differentiating again and transposing,

$$\frac{d^2\varphi}{d\theta^2} = \left(\frac{d\varphi}{d\theta}\right)\tan\varphi - \frac{\sin\theta}{n\cos\varphi}.$$

By substituting in this expression the above value of $d\varphi/d\theta$ and noting that $(n\cos\varphi)^2 = n^2 - \sin^2\theta$, we have

$$\frac{d^2\varphi}{d\theta^2} = (1-n^2)\frac{\sin\theta}{(n\cos\varphi)^3}.$$

Substituting this in the previously obtained value of d^2x/dt^2 and reducing, we have as the final value of the acceleration of the center of gravity in the direction of the engine axis,

$$\frac{d^2x}{dt^2} = r\omega^2\left\{\cos\theta + \left(\frac{b}{l}\right)\frac{n^2(\cos^2\theta - \sin^2\theta) + \sin^4\theta}{(n\cos\varphi)^3}\right\}.$$

9. We next obtain the acceleration of the center of gravity perpendicular to the engine axis. From inspection of Fig. 130,

$$y = \frac{a}{n}\sin\theta.$$

Differentiating and reducing,

$$\frac{d^2y}{dt^2} = -r\omega^2\frac{a}{l}\sin\theta.$$

10. Next we obtain the angular acceleration of the rod from the above expression for $d^2\varphi/d\theta^2$ giving

$$\frac{d^2\varphi}{dt^2} = \omega^2(1-n^2)\frac{\sin\theta}{(n\cos\varphi)^3}.$$

11. If we denote by X and Y the resultant forces acting on the rod in the direction of and perpendicular to the engine axis, and by M the sum of all of the force moments about the center of gravity, we have

$$X = \frac{Wr\omega^2}{g}\left\{\cos\theta + \left(\frac{b}{l}\right)\frac{n^2(\cos^2\theta - \sin^2\theta) + \sin^4\theta}{(n\cos\varphi)^3}\right\}$$

$$Y = -\frac{Wr\omega^2}{g}\frac{a}{l}\sin\theta$$

$$M = -\frac{Wk^2\omega^2}{g}(n^2-1)\frac{\sin\theta}{(n\cos\varphi)^3}.$$

Next we express X, Y and M in terms of the forces P, G and C, giving,

$$X = P + C\cos\theta$$
$$Y = G - C\sin\theta$$
$$M = Pa\sin\varphi - Ga\cos\varphi - Cb\sin(\varphi+\theta).$$

12. By substituting the values of P and G obtained from the two first equations in the third one and reducing we obtain

$$C = \frac{Xa\sin\varphi - Ya\cos\varphi - M}{l\sin(\varphi+\theta)}.$$

In this expression are substituted the values of X, Y and M in terms of θ, deduced above. The resulting expression is the value of C in terms of θ, which we are seeking. The expression as originally obtained is somewhat involved, but after considerable reduction assumes the comparatively simple form

$$C = \frac{Wr\omega^2}{g}\frac{a}{l}\left\{1 - V\left(\frac{b}{l} - \frac{k^2}{al}\right)\right\} \quad . \quad . \quad . \quad . \quad (1)$$

where V is a function of θ having the value given in the notation above. (1) is the final expression for the exact centrifugal force of the connecting-rod.

13. From the expression $P = X - C\cos\theta$, and the expression (1), we obtain the value for P in terms of θ, which reduces to

$$P = \frac{Wr\omega^2}{g} \times$$

$$\left[\left(1 - \frac{a}{l}\left\{1 - V\left(\frac{b}{l} - \frac{k^2}{al}\right)\right\}\right)\cos\theta + \left(\frac{b}{l}\right)\frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right]. \quad (2)$$

This is the final exact expression for the portion of the piston pressure absorbed by the rod inertia.

14. From the expression $G = Y + C\sin\theta$ and (1) we may obtain an expression for G, the guide pressure due to the rod inertia. This turns out to be very small for connecting rods of the usual type, and is quite negligible as compared with the thrust due to the obliquity of the rod. That is to say the effect of the rod inertia on the guide pressure is insignificant. Therefore we will not consider G any further.

Reduction of Expression for Centrifugal Force.

15. The expression (1) for the outward centrifugal force of the rod (that is the radial pressure between rod and crank-pin due to the rod inertia), may be simplified by making some slight approximations.

In the first place it is to be remarked that the centrifugal force of the rod is not constant since the expression (1) contains V, a function of θ. We have

$$V \equiv \frac{1 - 1/n^2}{\left(1 - \frac{\sin^2 \theta}{n^2}\right)^2 \left(1 + \frac{\cos \theta}{n\sqrt{1 - (\sin^2 \theta)/n^2}}\right)}.$$

Hence the centrifugal force of the rod varies somewhat as the crank rotates, so that it can never be exactly compensated for by a fixed counterbalance placed opposite the crank.

16. However, the variation of V is not very great. If n has the average value 5, V will vary from 0.8 to 1.2 for all values of θ. For average connecting rods the quantity $(b/l - k^2/al)$ is about $\frac{1}{9}$. Hence the variable term in the expression (1) for C varies from $(1 - 0.8/9)$ to $(1 - 1.2/9)$, or from 0.91 to 0.87, a variation of about 2 per cent. from the mean value. That is to say, for the average case taken, a counterbalance placed opposite the crank sufficient to balance the mean value of the centrifugal force of the rod will be about 2 per cent. too heavy at some crank positions and 2 per cent. too light at others. This variation is inappreciable and hence the centrifugal force of the rod can be properly balanced by a counterbalance placed opposite the crank.

17. For the usual values of n the expression $1 - (sin^2\theta)/n^2$ is not very different from unity, for all values of θ. Therefore, the only term which varies appreciably as θ varies, in the expression for V, is $\cos \theta$ in the denominator. Cos θ has its maximum and minimum values for $\theta = O$ or π, at which points $1 - (sin^2\theta)/n^2$ is exactly unity. Therefore, we have the maximum and minimum values of V almost exactly corresponding to the maximum and minimum values of cos θ. These extreme values of cos θ are + 1 and − 1, and then V is $1 + \frac{1}{n}$ and $1 - \frac{1}{n}$. Hence a mean between the greatest and least values of V is $V = 1$. Inserting this value in (1) we have

$$C = W\frac{a}{l}\left\{1 - \left(\frac{b}{l} - \frac{k^2}{al}\right)\right\}\frac{r\omega^2}{g}. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

which reduces * to

$$C = \frac{K^2}{l^2} W \frac{r\omega^2}{g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This is the final expression for the average value of the centrifugal force of the connecting rod. The extreme values as the crank rotates are a few per cent. above and below this, as already shown.

The centrifugal force of the rod is the same as the centrifugal force of a mass concentrated at the crank pin whose weight is the fraction K^2/l^2 of the total weight of the connecting rod. *Hence, if we assume that the fraction K^2/l^2 of the rod's weight is a rotating part concentrated at the crank pin and counterbalance it with the crank pin and other rotating parts, we exactly neutralize the average centrifugal force of the rod and eliminate its tendency to shake the engine bed.*

In a number of rods calculated by the writer this fraction came out about ½, and this may be assumed as the average value.† Therefore, in default of knowledge of the exact value of K^2/l^2 for a given case, one-half of the rod's weight should be considered as the rotating part. It is often stated that one-third or one-fourth of the rod's weight is the rotating part, but there is no proper basis for such statements, and the correct value is probably always very close to one-half.

18. Let us consider the form (3) for the expression of the centrifugal force of the rod. It will usually be found that the term $\left(\frac{b}{l} - \frac{k^2}{al}\right)$ is a small quantity. In the rods calculated by the writer, its value was in the neighborhood of 1/9. Hence the fraction of the rod to be considered as rotating is somewhat less than the fraction a/l. Now a/l is the fraction of the rod's weight which is taken by the crank pin. Hence the fraction of the rod's weight to be counted as a rotating part is somewhat less than the fraction supported by the crank pin, being about 8/9 of it in an average case. It is often stated that the portion of the rod to be counted as rotating should be exactly that portion supported by the crank pin, but this is not correct.

* This reduction, although perfectly obvious, was not made by the writer until Professor Mollier's paper was seen.

† Professor Mollier arrives at the same conclusion.

Reduction of Expression for Reciprocating Force.

19. Next let us consider the expression (2) given above for the reciprocating force P. The first term contains the coefficient

$$1 - \frac{a}{l}\left\{1 - V\left(\frac{b}{l} - \frac{k^2}{al}\right)\right\}.$$

As we have already remarked, V is a variable quantity, varying from 0.8 to 1.2 in an average case, and $\left(\frac{b}{l} - \frac{k^2}{al}\right)$ is a small quantity averaging about $\frac{1}{5}$. Hence the above coefficient has a value about the same as $1 - \frac{a}{l}$ or $\frac{b}{l}$, which is the coefficient of the second term of the expression (2) for P. Now it may be shown that the fraction comprising the second term of (2) is very small compared with the first term, $\cos\theta$.* We will make but slight error in the expression for P if we use an approximate value for the coefficient b/l of this small second term by taking it as the same as the above coefficient of the first term. Hence a very close approximate expression for the reciprocating force P is

$$P = \frac{Wra^2}{g}\left[1 - \frac{a}{l}\left\{1 - V\left(\frac{b}{l} - \frac{k^2}{al}\right)\right\}\right] \times \left[\cos\theta + \frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right]. \quad . \quad (5)$$

20. The second parenthesis in this expression will be recognized as being the variable factor in the exact expression for the force required to accelerate the actually reciprocating parts. (See any discussion of the Inertia of the Reciprocating Parts. In *Unwin's Machine Design*, Volume II., page 117, equation 4,† 1902 Edition.) The first parenthesis contains the variable factor V and therefore the reciprocating force required to accelerate the connecting-rod is not quite the same as if some fraction of the rod were a truly reciprocating part. However, just as in the case of the centrifugal force C, the variation of V as the crank rotates gives a variation in the value of P of about 2 per cent. on either side of its mean value.

* The fraction $\frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}$ may be written $\frac{\cos 2\theta + (\sin^4\theta)/n^2}{n\,(1 - (\sin^2\theta)/n^2)^{\frac{3}{2}}}$. Now $1/n^2$ is a small quantity, being about .04 in an average case, so that it may be neglected. The fraction then becomes $(\cos 2\theta)/n$, whose average value is small compared with the average value of $\cos\theta$.

† There is a typographical error in this equation in this edition. The sign in the numerator of the fraction should be + instead of − .

21. In view of the great simplicity which results if we are able to consider a fraction of the rod weight as a truly reciprocating part, it is permissible to neglect the variation of P due to this variation of V as the crank rotates. The mean value of P we find by inserting the mean value $V = 1$ in (5), giving after reduction,

$$P = W\left(1 - \frac{K^2}{l^2}\right)\frac{r\omega^2}{g}\left[\cos\theta + \frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right].$$

This is the final closely approximate value of the reciprocating force required for the acceleration of the connecting-rod and supplied by a portion of the piston pressure. As stated, the actual value of P is in an average case about 2 per cent. more and less than this value.

22. Our final approximate expression for P is the same as the force required to accelerate a purely reciprocating part whose weight is $W(1 - K^2/l^2)$. This it will be noticed is the remainder of the rod weight after the portion already shown to be the rotating part has been deducted. The reciprocating force required by the rod inertia is therefore the same as if the fraction of the rod weight given were a purely reciprocating mass, concentrated at the cross-head-pin or wrist-pin. *We may therefore almost exactly take account of the inertia of the rod by including the fraction* $(1 - K^2/l^2)$ *of its weight with the weight of the piston and cross-head, and finding the inertia forces for all together.* The fraction of the rod weight to be taken as a reciprocating part comes out about one-half in average cases. It is slightly more than b/l, the fraction of the rod's weight supported by the wrist-pin.

23. The inertia of the reciprocating parts is taken account of, as explained in treatises on engine design, by drawing a line on the indicator diagram to represent the force required for their acceleration. The distance between this line and the line giving the net pressure on the piston represents the effective force transmitted to the crank-pin.

Forces Shaking the Engine Bed.

24. As already stated, the radial or centrifugal force at the crank-pin due to the inertia of the rod is completely neutralized by counterbalancing the fraction K^2/l^2 of the rod's weight conidered as concentrated at the crank-pin. Of course the crank-pins

itself and all other rotating parts must also be exactly counterbalanced.

25. If this amount of counterbalance only is provided, none being added for the reciprocating parts, the engine will be exactly in equilibrium in the direction at right angles to its axis. That is to say, there will be no force tending to shake the engine bed perpendicular to the engine axis. It must be noted that in order to secure this result the fraction K^2/l^2 (which is usually about $\frac{1}{2}$) of the rod's weight must be counterbalanced. It is often stated that only the purely rotating parts should be counterbalanced in order to eliminate shaking forces perpendicular to the axis, but this is not sufficient.

25. If the rotating parts, including the proper fraction of the rod, are counterbalanced only, there will be a force in the line of the engine axis tending to shake the bed, due to the inertia of the reciprocating parts. Additional counterbalance is therefore usually provided to partially or wholly neutralize this shaking force in the direction of the axis, at the expense of introducing a new shaking force perpendicular to the axis. A mean is usually struck, and a portion only of the reciprocating parts are balanced, usually from $\frac{1}{4}$ to $\frac{3}{4}$, according to circumstances. This is, of course, in addition to the counterbalance for the rotating parts. There are then moderate forces tending to shake the engine bed, both in the direction of and perpendicular to the axis. The details of this matter are given in works on steam-engine design.

It is to be noted in computing the weight of the reciprocating parts in order to counterbalance them that the fraction $(1 - K^2/l^2)$ of the connecting rod weight (which is usually about $\frac{1}{2}$) is to be included. The whole of the rod weight is often counted with the reciprocating parts, but this is not correct.

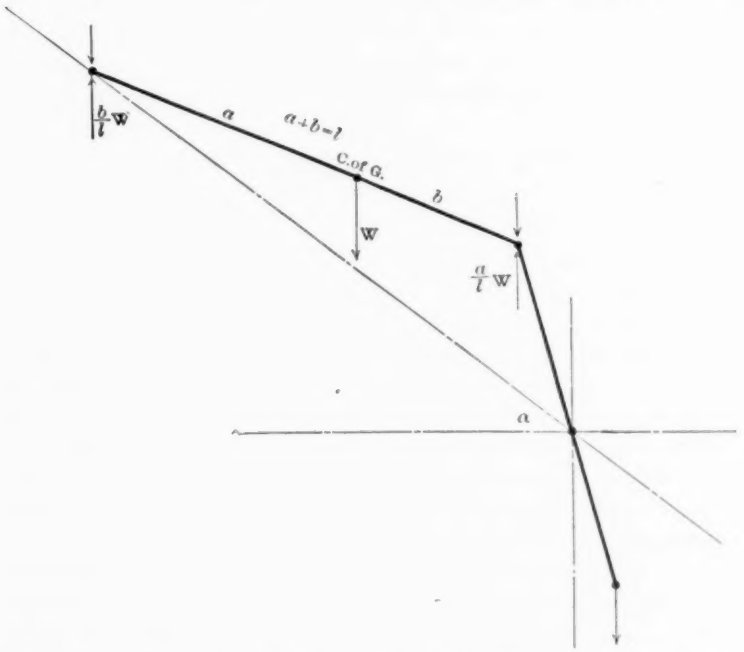
26. It may be remarked that there is an unbalanced turning moment on the crank-shaft due to the weight of the counterbalance provided for the selected fraction of the reciprocating parts in horizontal engines, and to the weight of the unbalanced portion of the reciprocating parts in vertical engines. For strict accuracy this effect should be taken into account in making crank-effort diagrams, etc. It is usually so small as to be negligible, however.

Effect of Connecting Rod Weight.

27. We have hitherto considered only the inertia effect due to the mass of the rod, and not the direct effect of its weight.

Let us consider the effect of the rod weight for the case of an engine whose axis is inclined at an angle α to the horizontal, as shown in Fig. 131. It will develop that α does not appear in the results at all, so that the effect of rod weight is exactly the same for a horizontal, vertical or inclined engine.

28. The rod weight is supported by vertically upward reactions at wrist-pin and crank-pin. These are found by considering the rod



Moss, S.A. DIAGRAM OF CONNECTING ROD WEIGHT. Am. Bk. Note Co., N.Y.

FIG. 131.

as a beam, loaded at the center of gravity and supported at the pins. The forces are all vertical and, hence, parallel, regardless of the inclination of the rod. By taking moments about each support successively, we arrive at values for the reactions which are constant regardless of the inclination of the rod, and are the same as if it were always horizontal. The reactions are Wa/l and Wb/l.

Of course, these reactions of the pins upon the rod have, as opposite aspects, vertically downward pressures of the rod upon the pins. There is, therefore, a pressure of the rod upon the crank-pin, always vertically downward, amounting to Wa/l. As we

have already shown, this is nearly the same, in an average case, as the fraction of the rod weight K^2/l^2 which must be considered as as a rotating part in order to take account of inertia. Hence we take exact account of the inertia of the rod, and approximate account of its weight, by supposing that the fraction K^2/l^2 is concentrated at the crank-pin.

29. The counterbalance provided for the crank-pin and other truly rotating parts, of course, neutralizes the effect of their weight in causing a turning moment on the crank-shaft. Now the additional counterbalance provided to neutralize the centrifugal force of the rod balances a fraction K^2/l^2 of the rod weight, while the weight of the rod on the crank-pin is the fraction a/l, which is nearly the same. Hence, in an average case, the weight of the counterbalance provided for the centrifugal force of the rod nearly neutralizes the effect of the weight of the rod on the crank-pin in tendency to cause turning moment on the crank-shaft. That is to say, the weight of the rotating parts, including the portion of the rod considered as rotating, and the weight of the counterbalances for them, need not be considered in making crank-effort diagrams. As already remarked, the weight of the counterbalance for the balanced fraction of the reciprocating parts gives a slight unbalanced turning moment on the crank-shaft in horizontal engines.

30. The load on the wrist-pin due to the rod weight is a constant vertically downward force amounting to Wb/l. The fraction b/l of the rod weight must therefore be added to the weight of the cross-head. As we have already shown, this is nearly the same, in an average case, as the fraction of the rod weight $(1 - K^2/l^2)$ which must be considered as a reciprocating part in order to take account of inertia. Hence we take exact account of the inertia of the rod and approximate account of its weight by supposing that the fraction $(1 - K^2/l^2)$ is concentrated at the wrist-pin. In order to take exact account of the weight the fraction should be b/l.

31. In both horizontal and vertical engines there is a guide pressure due to connecting-rod thrust which is fully discussed in works on engine design. In horizontal engines there is an additional guide pressure due to the weight of the cross-head plus the fraction of the connecting-rod referred to above.

31. In vertical engines the total weight of the reciprocating parts, including the fraction of the connecting-rod above referred to, gives a turning moment on the crank-shaft. The action is the

same as if the piston pressure were increased during the down stroke and decreased during the up stroke. If counterbalance is provided for a portion of the reciprocating parts, a portion of this effect is neutralized. That is to say, only the weight of that fraction of the reciprocating parts not counterbalanced gives a turning moment on the crank-shaft in vertical engines.

Summary of Results.

32. l = distance from center to center of connecting-rod.
a = " " " of gravity of rod to wrist-pin.
b = " " " " " " " " crank-pin.
K = radius of gyration of rod about an axis through the wrist-pin.

A fraction of the connecting-rod weight

$$\frac{K^2}{l^2}$$

is to be counted as a rotating part, concentrated at the crank-pin, and counterbalanced with the other rotating parts. When this is done there is no shaking force on the engine bed due to the inertia effect of the rod on the crank-pin, and practically no turning moment on the crank-shaft due to the weight of the rod on the crank pin. The given fraction of the rod comes out nearly one-half in ordinary cases, and this value may be assumed if the radius of gyration is not known.

In an average case the given fraction of the rod is about eight-ninths of the portion of the rod whose weight is supported by the crank-pin, a/l.

33. The remainder of the rod weight after the fraction above given as a rotating part has been taken out, that is, the fraction

$$1 - \frac{K^2}{l^2}$$

is to be counted as a reciprocating part concentrated at the cross-head-pin or wrist-pin. When this is done, the usual methods of computing the counterbalance and the forces required to accelerate the reciprocating parts will give correct results. The given fraction of the rod comes out nearly one-half in ordinary cases and this value may be assumed if the radius of gyration is not known.

In an average case the given fraction of the rod is about one

and one-eighth of the portion of the rod whose weight is supported by the wrist-pin, b/l.

The total cross-head weight pressing on the guides in horizontal engines, or the downward force due to the weight of the reciprocating parts in vertical engines is to be found by including the fraction b/l of the connecting-rod weight. However, in most cases, it will be sufficiently accurate to include the same fraction of the connecting-rod as is used for the inertia effect.

DISCUSSION.

Mr. G. R. Henderson.—There is one portion of this paper which possibly I do not correctly understand. Paragraph 22 states:

"We may, therefore, almost exactly take account of the inertia of the rod."

(He is now referring to the reciprocating inertia.)

"By including the fraction $(1 - K^2/l^2)$ of its weight with the weight of the piston and crosshead, and finding the inertia forces for all together."

This would seem to indicate that only a portion (about one-half) of the rod is assumed as the reciprocating weight. If it be assumed that one-half is the revolving weight, it is perfectly proper in balancing to allow for one-half, but the true reciprocating force is due, of course, to the whole weight of the rod. It is not stated very clearly in this as to whether that has been so considered or not. I merely wish to call attention to that because there is no doubt that the whole weight of the rod must be considered as the reciprocating weight. If part of it has been balanced for revolving weight, then that portion should be omitted in the balancing for the rest of the reciprocating weights.

*Mr. S. A. Moss.**—In paragraph 22 and again in paragraph 33, I have endeavored to state as clearly as I knew how, that about *one-half* of the rod is to be counted as a reciprocating part, and not the whole rod. Mr. Henderson states, "there is no doubt that the whole weight of the rod must be considered as the reciprocating weight." This statement is totally erroneous. This statement is made in several text-books, without proof, and is regarded as correct by many engine builders. Nevertheless, it is incorrect, as is shown mathematically in the present paper. In

* Author's Closure, under the Rules.

other words, a great part of the present paper is devoted to a definite proof that the whole weight of the rod must not be considered as a reciprocating part, since about half of the rod is as truly rotating as the crank pin itself.

If the weight of the crank pin be increased by about one-half of the rod-weight, and if the weight of the cross-head pin be increased by the balance of the rod-weight, and if the rod then be regarded as wholly without mass, the inertia of the rod will be correctly accounted for. It is, of course, understood that the correct value of the fraction to be counted as the reciprocating part is $(1-K^2/l^2)$, and that the value $\frac{1}{2}$ is only an approximation, correct for an average case, and to be used when the radius of gyration of the rod is not known.

No. 1072.*

CONDENSERS FOR STEAM TURBINES.

BY GEORGE I. ROCKWOOD, WORCESTER, MASS.
(Member of the Society.)

1. Visitors to steam turbine power stations generally notice, as a first impression, the relative inconspicuousness of the turbine itself in the midst of its numerous and large condenser auxiliaries. This prominence of the condenser equipment is especially noticeable in the case of the smaller turbine plants, where the floor space occupied by the condenser system far exceeds that required by the turbine.

2. There appear to be three accepted designs for these condenser systems: First may be mentioned the combination consisting of a surface condenser, a centrifugal hot-well pump, an air cooler, a single-cylinder dry vacuum pump, a centrifugal circulating-water pump, together with their connecting and drip piping and valves. Second, the foregoing arrangement may be varied by omitting the hot-well pump and also the air cooler and dry vacuum pump, substituting the wet vacuum pump—preferably of the Edwards type. The cost per kilowatt is about the same—$7 to $10—for either of these systems. The third system is like the first, except that an elevated jet condenser with barometric tube and hot-well take the place of the surface condenser and hot-well pump. The advantage of this third type over the surface condenser systems is that it may take up less floor space, while its cost is but $5 to $6 per kilowatt. The dry vacuum pump used with barometric condensers must have a two-cylinder air-pump, and the exhaust steam from the turbine cannot be used again in the boilers.

3. The whole cost of a condenser system, as intimated in the opening paragraph, is not, however, fully expressed in the con-

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

tract with the condenser contractor. The turbine room is made a full third larger, and hence more expensive, by the provision of the necessary floor space; space, too, that is needed in the operations of the turbines. The smaller the turbine plant is, the greater this part of the cost becomes relative to that of the whole power plant.

4. There is, besides, a fourth type of condenser which may be used with steam turbines and to which it is the object of this paper to direct attention; namely, the injector or ejector condenser.

5. Within the past year the Atlantic Mills, Providence, R. I., have installed a 400 kilowatt Westinghouse-Parsons turbine. A vacuum of 28 inches to 28½ inches is maintained on this turbine by means of the following condenser system: The exhaust steam is led, through twenty feet of vertical 16-inch cast iron pipe and three short-turn elbows, into a 16-inch Bulkley injector condenser. The level of the ground floor is thirty-four feet below the condenser bulb, while the turbine lies on a concrete-steel floor, the level of which is twelve feet and six inches above the ground floor.

6. The injection water comes 500 feet from the river to the power house, under a slight head—perhaps three feet—depending upon the state of the river. A 6-inch belt-driven Lawrence centrifugal pump elevates the water into a vertical tank, 30 inches by 15 feet deep. The level of the water in this tank is maintained by the waste pipe six feet below the water inlet nozzle on the condenser. From near the bottom of this vertical tank a 7-inch injection pipe rises up to the condenser.

7. A good deal of air along with the water is pumped into the top of the tank by the centrifugal pump, but apparently the depth of the tank acts efficiently as an air separator, and no air in the form of bubbles passes over into the condenser. At any rate, the vacuum shown by the mercury column is 28½ inches. The remarkable thing is that this is so, as nearly as the height of a mercury column can be measured, whether the column be attached to the bulb of the condenser or to the exhaust chest of the turbine.

8. To prove this absence of friction in the exhaust pipe a ¼-inch pipe was connected to the elbow above the condenser, then led down and attached to the turbine exhaust chamber. A branch horizontal pipe leads off to a mercury column. Two valves, one above and one below this horizontal pipe, enable either connection to be made at will. Starting with both valves open and the mercury quiescent, it was impossible to perceive any move-

ment of the mercury, either up or down, after shutting either valve.

9. Not only is a 16-inch exhaust pipe thus proved ample and more than ample in size for a 400 kilowatt turbine, but the vacuum obtainable with this condenser is substantially the same, whether steam is passing through the turbine or not. The falling of the water through the " throat " is the air pump, and is the only air pump needed. The builders provided a 20-inch exhaust nozzle on the turbine, but Mr. Bulkley believes that, on the contrary, a 14-inch condenser would do as well as the 16-inch.

It may be of interest to relate here a rather unusual experience, encountered when this 16-inch condenser was first put into service.

10. It was convenient to have the injection pipe rise on that side of the 16-inch exhaust pipe farthest from the condenser. At the top a 45 degree bend connected to a long-radius elbow enabled the pipe to turn and pass the exhaust pipe, approaching the condenser horizontally instead of vertically as in the usual case. Upon starting up the turbine only twenty-two inches of vacuum could be obtained. The piping was, nevertheless, proved to be absolutely tight. After shutting the injection valve Saturday afternoon with the turbine blanked off at its nozzle, and with 22 inches vacuum on the exhaust pipe and condenser, one found 12 inches vacuum still left on the system Monday morning before removing the blank flange in preparation for starting. Still, only 22 inches was the maximum height of the mercury column while running.

11. Finally, at the suggestion of Mr. Bulkley to the effect that in his experience he had found it necessary to have the injection pipe to his condensers approach the condenser vertically rather than horizontally for the best results (although he had no explanation satisfactory to himself why this was so), a re-arrangement of the injection piping was made, permitting this vertical approach. At once, upon starting up the turbine, the result was 28½ inches of vacuum, and this has been maintained without interruption since. With the present load—about 300 kilowatts to 350 kilowatts—the 7-inch injection valve is open only a few turns, the temperature of the hot-well is from 80 degrees to 90 degrees, and water enough can be passed through this condenser to maintain the vacuum in summer weather. The highest degree of vacuum thus far recorded, as measured in the turbine exhaust chamber, is 28¾ inches, the lowest 27½ inches. The variation is caused

partly by changes in the barometer, but more by variable leakages in drip-valve seats. Perhaps also the amount of air entrained by the injection water varies from time to time.

12. If water is not to be had in abundance, then the best way is to have two pumps, one of which returns to the condenser, water taken from the hot-well. In this way the highest theoretical temperature of the hot-well water may be reached. It takes 10 horsepower to supply water by means of the centrifugal pump in sufficient quantity to condense 400 kilowatts of steam.

The method of sealing the spindle of the Parsons turbine against air leak where it passes out to its journals from the low-pressure chambers, namely, by pumping water with centrifugal pumps formed in small recesses in the shaft cover, so as to keep a water pressure in these recesses in excess of that of the atmosphere, is a perfect success, as the experience with this condenser shows, although when but 22 inches could be obtained, owing to the fault in the injection pipe design described, the efficacy of these air seals was very seriously doubted.

It is interesting to note our experience that no drip pipe or drip pump, for removing the water of condensation or the leakage into the exhaust chamber from these air seals, is found to be necessary or desirable.

13. It is, indeed, necessary to exclude any accumulation of water in the exhaust pipe for fear that it will sway back and forth until it flushes up on to the large low-pressure blades of the turbine. Running as they do at a very high velocity, sudden contact with water from the exhaust pipe will strip the last row off clean if such contact is permitted. Any further damage, however, to the other rotating blades seems to be prevented by the presence of the fixed row, which, by dividing up the water into small streams, seems to protect the moving rows from contact with solid water and therefore from injury.

14. The rate of accumulation of water leaking by the small centrifugal air-seal pumps into the exhaust pipe may be any amount up to over 950 pounds an hour. If, however, there is the least external load on the turbine, the flow of the steam up the exhaust pipe has the power to sweep the exhaust chamber dry. The method of starting the turbine at the Atlantic Mills is to first turn on the injection water. Then admit steam to the turbine without admitting water to the air seals. After the load begins to come on, as shown by the ampere meter, the drip-pipe from

the exhaust chamber to the atmosphere, which has been open all night and up to this point, is now closed, the water turned on to the air seals, and the vacuum immediately draws down to 28 inches. It is not found absolutely necessary to start in this way, as the turbine can be run hours before the water accumulates in quantity. The object is simply to drain the turbine up to the moment when the load begins to come on.

15. If the load is a "jumping" one, from nothing to full load, there is no danger of water accumulation. It is, of course, quite possible to provide a drain receiver and pump which will work under the vacuum if it is deemed desirable to do it. There is no exhaust steam "entrainer," or water trap or seal, provided at the Atlantic Mills turbine.

16. The injector condenser costs, fully installed, with centrifugal circulating water pump, tank piping and valves, from $2 to $2.50 per kilowatt. This cost is much reduced if there is a natural head of water available. At the installation described the condenser, two elbows, one tee, 18 feet of cast-iron exhaust pipe and a 14-inch automatic relief valve cost, erected, $591.50. The room it occupies is practically none at all.

17. Theoretically considered, in its relation to turbines, the injector condenser would seem to bar out all other condenser systems in those situations where the water used in the boilers is pure in its natural state. Where it is absolutely essential to save the water of condensation for re-use as boiler feed-water, owing to the presence of salt, sulphate of lime, wool grease, acid from steel works, or other hurtful pollution, then it pays to use one of the surface condenser systems. If the city water is pure and costs about seven cents per 1,000 gallons, water enough for a 400-kilowatt machine at 100 per cent. load factor would cost per factory year of 310 days, 10 hours a day, at 1,000 gallons per hour, about $217. With interest at 5 per cent. and fixed charges at 8 per cent., this sum warrants the capital expenditure of not to exceed $1,670. It is thus clear that it does not pay to buy the surface condenser system simply to save the cost of paying city rates for boiler feed-water.

18. Much talk has been made about the freedom of the condensed exhaust steam in turbines from cylinder oil, and the advantage which this purity gives to it as compared with the oily exhaust from reciprocating engines, where the condensed steam is returned to the boilers.

19. It should not be forgotten that great purity of feed-water is not in itself a desirable thing, being only better than very impure water; for it pits the tubes and water-legs of steel boilers unless some lime is added.

20. Where the waste injection water from a jet condenser is used for washing in a dye house, this freedom from oil is of great advantage. Such is the aversion of dyers to using exhaust steam to heat water on account of the supposed presence of cylinder oil, no matter how careful one may be to provide and operate successfully oil eliminators, that this freedom of the turbine from the use of cylinder oil is sufficient cause to determine the purchase of turbines in place of engines for power in such places.

21. The Atlantic Mills turbine is, so far as the writer knows, the only instance on record of the use of injector condensers for large turbines either of the Parsons or Curtis type.

DISCUSSION.

Mr. Jos. Morgan.—Mr. Rockwood's paper on "Condensers for Turbines" contains valuable information needed by engineers contemplating the use of new apparatus of this kind. The writer has had experience in the use of barometric condensers, and also has had some recent experience in the use of Parsons turbines at the Broad Street plant of the Citizens' Electric Co. in this city, where we have tested the actual working results of a Weiss condensing plant in connection with the turbines. These results have been furnished to Mr. Bibbins, an engineer of the Westinghouse Co., who examined the plant on behalf of the turbine builders, and is about to publish a discussion of the results in the January number of the periodical "Power." I send you a plan showing the general layout of the Citizens' Co. boilers, turbines and condensing plant, with their pipe connections.

This plant has also been described in "Electrical World," September 3d, and last September issue of "Power," and will include, when completed according to the plans, three 400 kilowatt Westinghouse-Parsons turbines, two of which are now installed, all exhausting into a Weiss condenser sufficient for 1,200 kilowatts of turbines, with air and water pumps in two independent units, each of which is sufficient to carry for a time two of the turbines with an overload of 50 per cent. Under

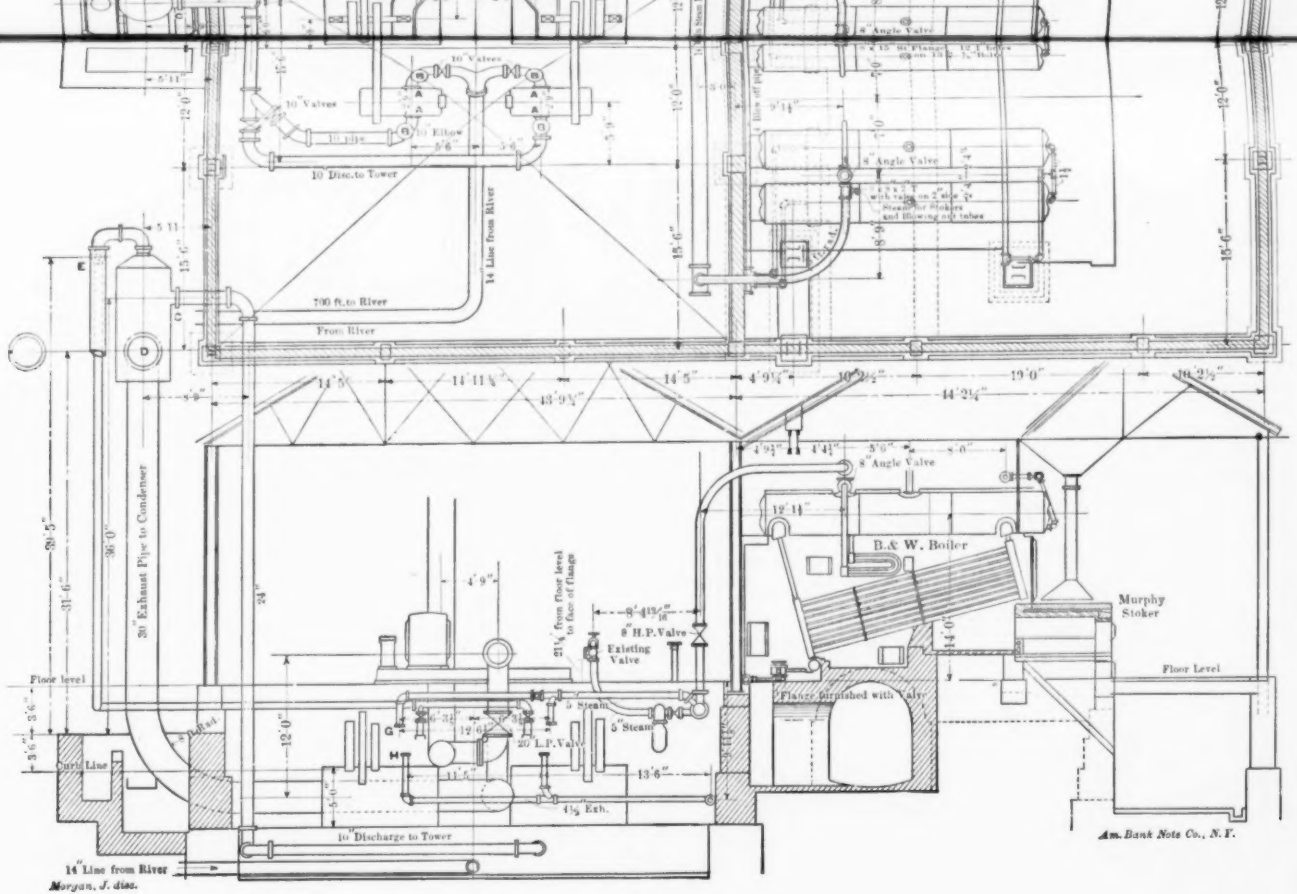
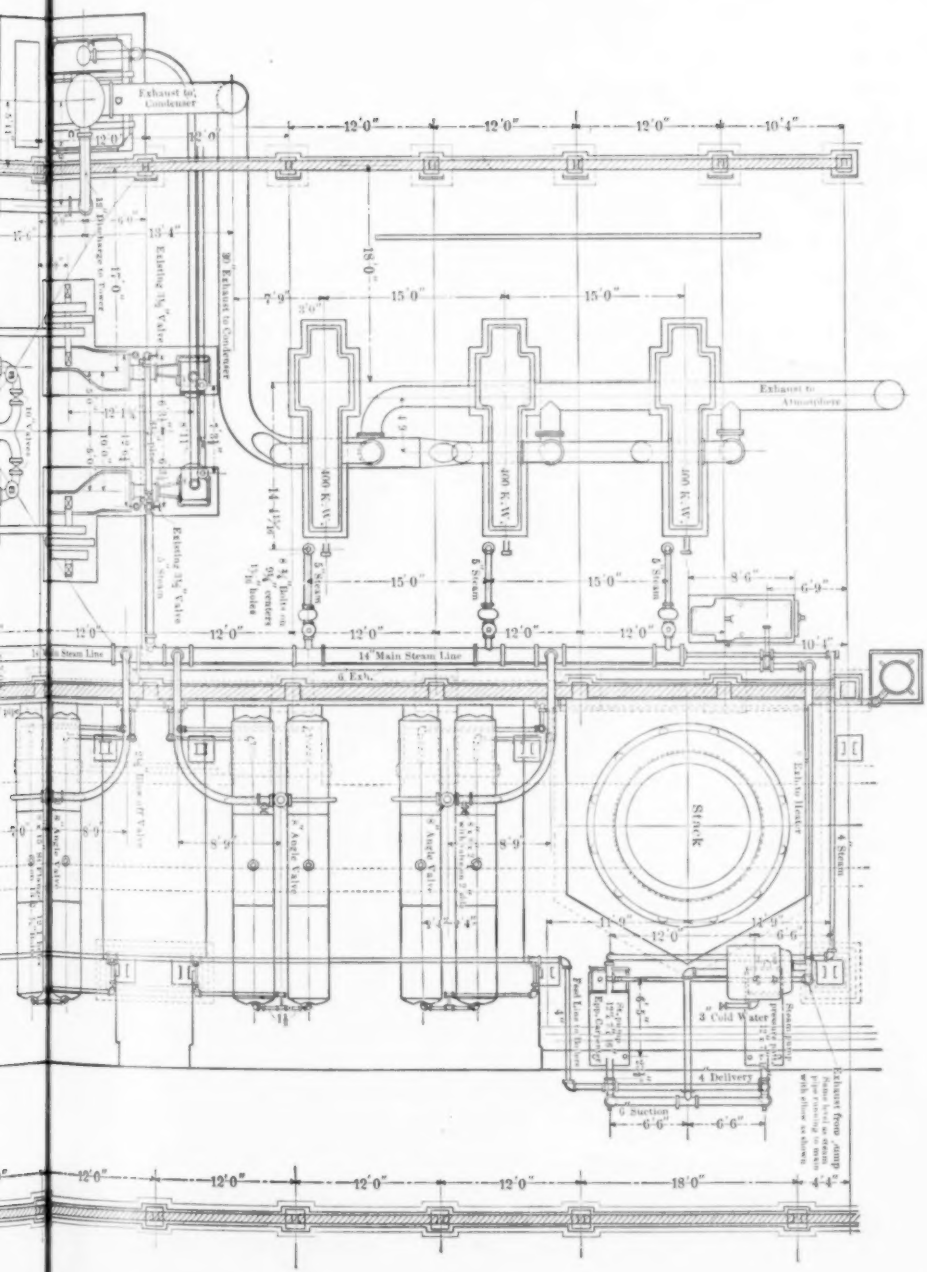

10" Valves
10" Valves
10" Elbow
10" Disc. to Tower
14" Line from River
700 ft. to River
From River
8" Angle Valve
8" Angle Valve
Steam for Stokers and Blowing out tubes
8" Angle Valve
B. & W. Boiler
Murphy Stoker
Floor Level
Flange furnished with Valve
Floor level
30" Exhaust Pipe to Condenser
8" H.P. Valve
Existing Valve
21¼" from floor level to face of flange
5" Steam
5" Steam
20" L.P. Valve
4½ Exh.
10" Discharge to Tower
14" Line from River
Morgan, J. disc.
Am. Bank Note Co., N.Y.

FIG. 132.


Exhaust to Condenser
12'0"
12'0"
12'0"
10'4"
18'0"
15'0"
15'0"
Exhaust to Atmosphere
400 K.W.
400 K.W.
400 K.W.
14" Main Steam Line
6" Exh.
Stack
Exhaust Header
4" Steam
3" Cold Water
4" Delivery
6" Suction
Exhaust from Pump
12'0"
12'0"
12'0"
12'0"
18'0"

ordinary conditions of load factor, one pumping plant is more than ample to take care of the generating plant with its commercial load. The aim has been to amply guarantee the running of the electric plant by having a spare pumping plant. First cost was a consideration secondary to reliability.

In selecting this condensing plant, both the surface and Weiss condensers were considered, and also the cost of operation on the assumption that 50 pounds of water for the surface condenser and 30 pounds for the Weiss condenser per pound of steam were the relative proportions of water required for the same resulting vacuum, and that feed water would have to be bought at seven cents per 1,000 gallons in the second case. Considering the interest on first cost, the cost of running, and the cost of feed water, the Weiss was adopted as the cheapest. The working result is over 27 inches of vacuum with 12 indicated horsepower in the pump engine at 400 kilowatts on the generator, and deducting steam utilized in heating the feed water, this result is obtained by the expenditure for air and water pumps of about $2\frac{1}{2}$ per cent. of the total steam generated.

The condensing water is drawn from the Conemaugh River through a suction pipe over 700 feet long, and the static suction head is about 10 feet. When the condenser is working, the actual lift made by the pump is about 26 feet, the vacuum assisting the pump. The water in the river is unfit for boiler use, and the pure city water is run through the oil cooling coils in the turbine, and thence into the feed water heater. The use of the oil cooling water was an after-thought and an economy of feed water cost. It was not desirable to put the river water through the cooling pipes, as it is at times corrosive, having a considerable percentage of sulphuric acid due to mine drainage. The exhaust steam from the pump engine is used in the feed heater, as is the feed pump exhaust, thus providing warm feed even when other auxiliaries are stopped. The air pump is a single cylinder of the Weiss type arranged tandem with the steam cylinder of an engine driving by belt a Bibus rotary pump. The engine is controlled by a throttling governor driven by a hand adjustable friction gear so that the speed can be regulated to give the necessary fixed amount of water.

In this arrangement the condenser unit is not dependent upon the electric unit for its motive power, and the positive water pump being driven by a steam-engine at a governed speed, in

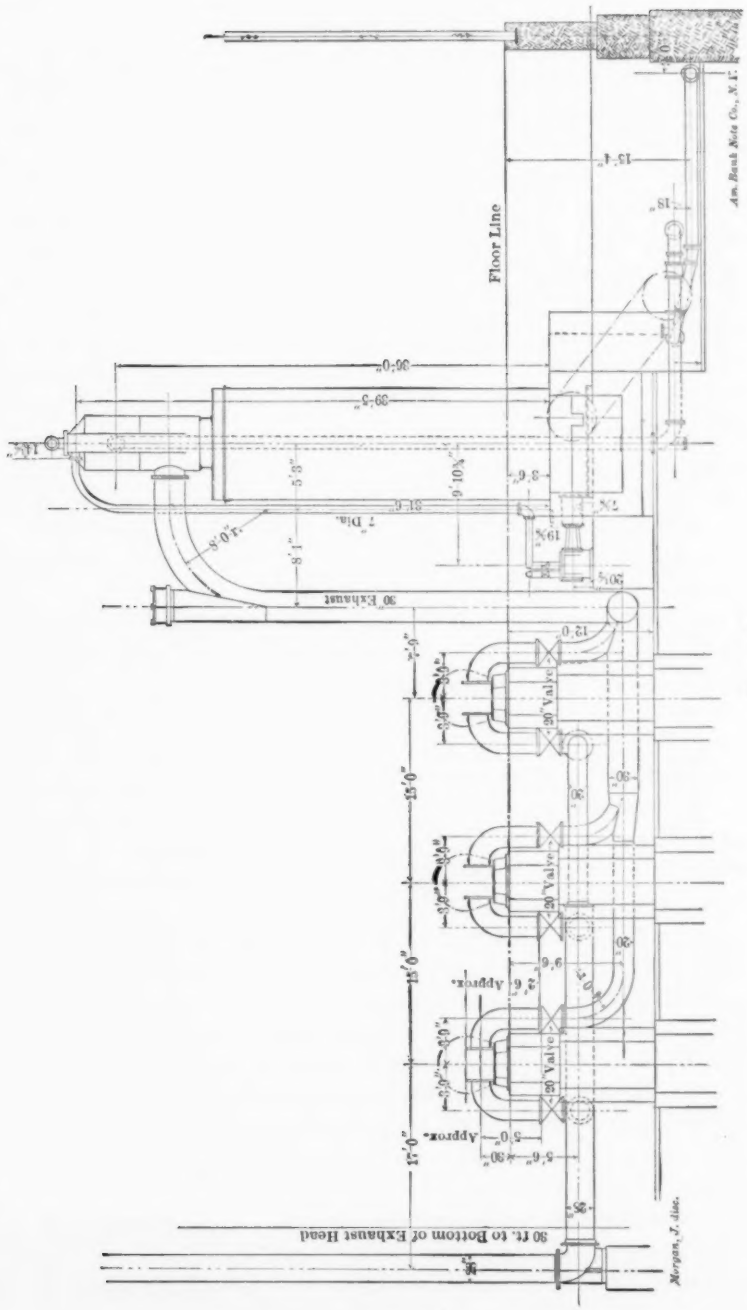

Floor Line
Am. Bank Note Co., N.Y.
36'0"
39'3"
5'8"
9'10½"
3'6"
31'6"
7" Dia.
3'0"
3'8"
30" Exhaust
12'0"
30" Valve
15'0"
9'6"
3'6" Approx.
5'0" Approx.
5'6"
30"
30 ft. to Bottom of Exhaust Head
Morgan, J. dec.

FIG. 133.

case of a very sudden increase of electric load, the condenser still receives the same quantity of water; and the vacuum, while temporarily lowered, is not lost as it would be in such a condensing plant as Mr. Rockwood describes, where the water flow is absolutely dependent upon the vacuum. In such a jet condenser, having no large condensing chamber, unless the quantity of water is greatly in excess of the average steam requirements, with any considerable increase of load, the condenser at once blows off, and the main engine labors along under non-condensing conditions or stops.

In a steam turbine unit under a jumping load, a jet condenser, therefore, means continual waste of water or occasional disastrous loss of vacuum. An electric lighting plant is, of course, more easy to handle than an electric power plant, as the fluctuations of load are more gradual and can be met by the necessary adjustments of water supply to the condenser. On our Johnstown City Lighting Plants we are still able to divide the loads among our various generators by hand adjustments of field rheostats, but in our Cambria Steel Works power plants we have on a 1,200 kilowatt plant frequent fluctuations of 600 kilowatts, lasting a few seconds or a few minutes, to which the steam governors must respond, and with variations in steam consumption which any satisfactory condensing plant must take care of.

I doubt the ability of such a plant as Mr. Rockwood describes to meet such conditions. We have had in the Cambria Works considerable experience with such jet condensers on automatic rolling mill engines. We were annoyed by their blowing off under sudden loads, and also found them wasteful of water, so that they have been replaced by several central condensing plants of the Weiss type. In some of these Weiss units we have arranged to take as much as 10,000 horse-power and the steam from ten large engines into one condenser. Owing to the extent of the piping systems, the large cubic capacity of the exhaust space and the great body of water always in the condensers, the load fluctuations produce relatively small and temporary effects. We have carried the steam in some cases through 1,500 feet of piping and have noted, as Mr. Rockwood has, the very small pipe friction. The vapor in the pipes under vacuum is so attenuated that its velocity is exceedingly great. We carry the exhaust steam, for instance, from our 1,200 kilowatt electric plant through 950 feet of 24-inch pipe, and a total distance of 1,000 feet, to a central

plant at the blast-furnaces. This is about 50 per cent. more steam than we had originally intended to put through this pipe, but under extreme fluctuations we drop momentarily less than 2 inches in vacuum. In another plant we carry the exhaust steam for a 500 horse-power rope haulage compound engine through 1,000 feet of 14-inch pipe to a large exhaust main, and thence 500 feet farther to the main condenser, with excellent results.

I can testify to the very satisfactory result of the water packing on the turbine shafts, as mentioned by Mr. Rockwood. It has been applied to the two turbines at the Citizens' plant, replacing the original steam packing with more satisfactory results as to tightness and saving of oil.

Mr. Francis Hodgkinson.—The vacuum obtained by Mr. Rockwood with the use of an ejector condenser, without the use of a separate air-pump, seems very remarkable. I should think, however, a very small air leak would seriously impair its vacuum. Mr. Rockwood bears witness to having a particularly tight piping system.

The installing of an air-pump would make this installation closely resemble the well-known "Barometric Condenser," and would seem to me to be an insurance of at least a fair vacuum should any of the apparatus be deranged or small air leaks develop, and would add practically nothing to the operating expense so long as no big air leaks exist.

I desire, however, to call particular attention to the 16-inch exhaust line, 20 feet in length and having three short turn elbows over which no drop of pressure could be observed. The Westinghouse-Parsons turbine, which exhausts into this condenser, showed the following results in a test, when operating with 28-inch vacuum, 150 pound brake horse-power, and 100 degrees superheat.

Brake H. P.	Total lb. steam condensed.	Lb. steam per brake H. P. hour.
588	7639.5	12.99
419.8	5759	13.71

When delivering 350 kilowatts (the load quoted by Mr. Rockwood) the steam consumption would closely approximate 6,767 pounds per hour, but as superheat is not being used at the present time, the consumption of steam would be within a very few per cent. of 7,520 pounds per hour.

With this consumption, the dryness factor of the exhaust steam

would be 0.908, and disregarding the water, the velocity in the pipe is 454 feet per second.

Supposing the exhaust pipe to be a well rounded orifice of 100 per cent. co-efficient and assuming the steam to be starting from rest, there would be a difference of pressure of 0.136 inches of mercury when passing this quantity of steam. Of course, a well-rounded orifice as above is assumed to be entirely without friction, and is a very different thing to 20 feet of pipe and three short elbows. I would have expected in the case of the exhaust pipe a back pressure of at least half an inch of mercury with the same quantity of steam.

It is very easy to be deceived in reading low pressures. A few days ago I had occasion to see if there existed a difference of pressure between two exhaust chambers of a turbine. I rigged up a manometer tube containing water, and in order to eliminate loops which might fill with water, and that the connecting pipes would freely drain, I was careful to elevate it so that the pipes connected to each side would have a continuous inclination downwards to the chambers whose pressure I desired to compare. I used ¼-inch pipe, the same size as that used by Mr. Rockwood.

The observed pressures were sometimes one side and sometimes the other, and frequently differences of pressure were shown that could not possibly exist. Quickly disconnecting the pipes showed that one of them contained a quantity of water which had condensed there, and I suppose because of capillarity and the smallness of the pipe would not drain away.

I then placed a small reservoir immediately at each side of the manometer and connected both top and bottom of each reservoir to each respective exhaust chamber, and was then able to keep the connecting pipes drained, and, I believe, obtained correct pressure readings.

Some time ago I had occasion to observe drops of pressure over an exhaust pipe during some tests with varying quantities of steam and vacua. The following list comprises the parts of the exhaust pipe described in their order.

1. 32-inch nozzle on the turbine.
2. 32-inch to 36-inch reducer, 16½ inches long.
3. Short turn 36-inch elbow.
4. Twenty feet of 36-inch pipe.
5. 36-inch valve (in no place is the area less than that of a 36-inch pipe).

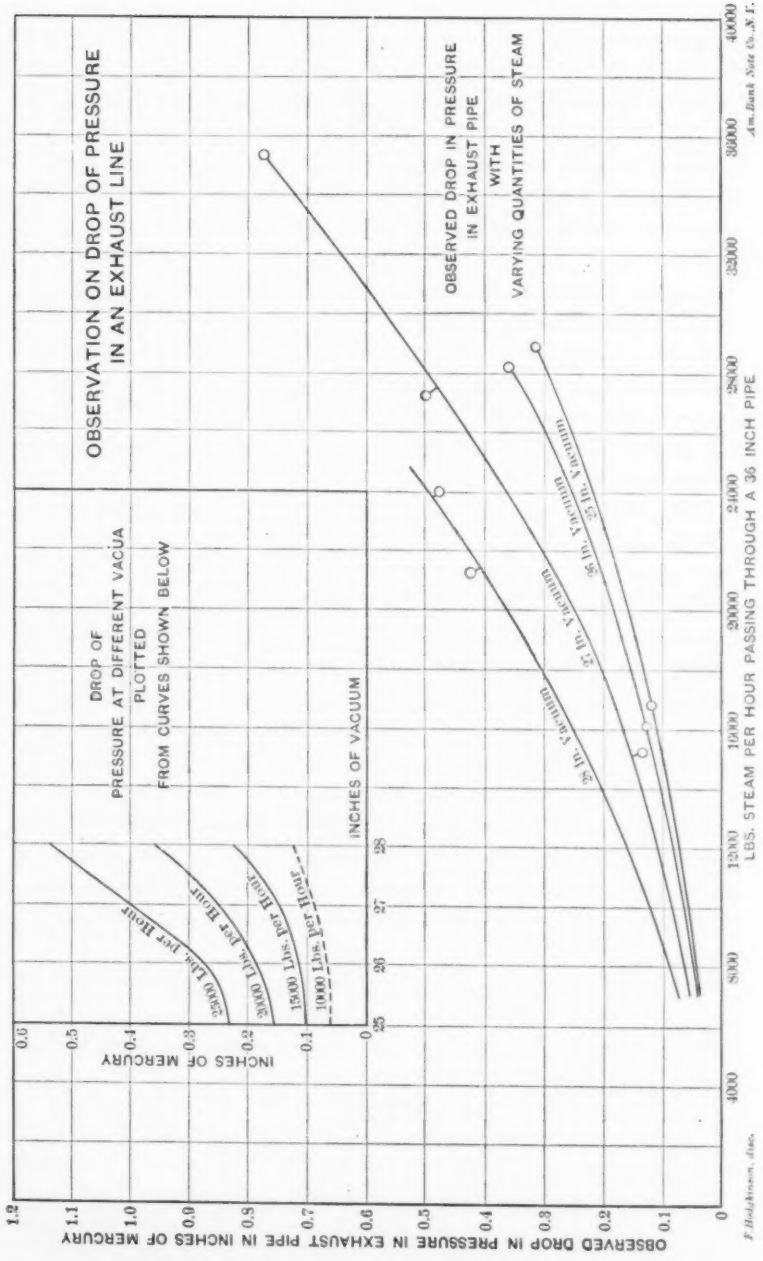

OBSERVATION ON DROP OF PRESSURE IN AN EXHAUST LINE
DROP OF PRESSURE AT DIFFERENT VACUA PLOTTED FROM CURVES SHOWN BELOW
OBSERVED DROP IN PRESSURE IN EXHAUST PIPE WITH VARYING QUANTITIES OF STEAM
INCHES OF VACUUM
INCHES OF MERCURY
OBSERVED DROP IN PRESSURE IN EXHAUST PIPE IN INCHES OF MERCURY
LBS. STEAM PER HOUR PASSING THROUGH A 36 INCH PIPE
2500 Lbs. per Hour
2000 Lbs. per Hour
1500 Lbs. per Hour
1000 Lbs. per Hour
25 in. Vacuum
26 in. Vacuum
27 in. Vacuum
28 in. Vacuum

Fig. 134.

6. 36-inch Tee piece.
7. Twelve feet of 36-inch pipe.
8. 36-inch Tee piece.
9. 36-inch Gate valve.
10. Five feet of 36-inch pipe.

A manometer connected to each end of this exhaust line gave the back pressure readings shown in the following table. In the same table are given the calculated velocities of steam through the pipe.

Vacuum at turbine exhaust referred to 30-in. barometer.	Lb. steam per hour from tests.	Observed drops in exhaust pipe in inches water.	Drops in exhaust pipe, inches of mercury.	Calculated velocity in 36-in. pipe assuming 90% quality of exhaust, feet per second.
25.05	16,809	1⅜	.12	85.8
25.01	28,902	4½	.312	146.1
26.05	16,106	1¾	.129	101.5
26.01	28,236	4⅝	.3585	176.5
26.95	15,241	1½	.138	123.6
27.0	21,189	3¼	.276	174
27.0	27,248	6¼	.496	223
26.89	35,297	10½	.773	279.2
28.0	21,286	5¼	.423	256.0
27.9	24,025	6½	.478	264.8

The manometer was connected by means of ¼-inch pipe, and there may have been some errors, such as those just described, but this is improbable, as it will be observed all the readings are fairly logical and plot into reasonably smooth curves as is shown in the accompanying diagram.

The exhaust line was certainly tortuous; nevertheless, it seems indicated that sizes of exhaust pipe required by the various turbine builders should not be materially cut down.

Mr. Frank B. Perry.—The fact that steam turbines are theoretically capable of utilizing the lowest available expansion of steam has undoubtedly caused manufacturers to advocate the type of condensing apparatus which will ensure and maintain the highest attainable vacuum. Condenser builders contend that a better vacuum can be obtained through the medium of a surface condenser equipped with either a wet or dry vacuum air pump and a closed hot-well system than is possible with a jet or an ejector condenser. The reason for this claim, apparently, is due to the

resultant effect of the liberal area of cooling surface provided combined with the efficiency of the vacuum pump in removing air from the entire system, including that contained in the feed water.

Since steam and water are intimately in contact with the injector condenser it would appear, theoretically at least, that the temperature and the quantity of cooling water may be varied within wider limits for procuring any desired vacua than is possible with a surface condenser. Practically, however, this consideration is of very little importance.

The introduction of the steam turbine brought the surface condenser more prominently into use for stationary engineering work as is evidenced by the equipments of nearly all the recent power plant installations. This type of condenser has been used extensively in marine practice and the difficulties at first encountered are now well known and have been almost entirely obviated by the substitution of mineral oils, sparingly circulated for internal cylinder lubrication, in place of the vegetable oils formerly utilized.

The exhaust steam from most turbines is free from oil. This is advantageous to the life of boilers, as no oil can be carried over with the condensed steam used for feed water to form scales or other injurious deposits on the plates. The pitting effect that is apt to be caused by the action of distilled water on boilers may be prevented with reasonable certainty, and, therefore, is a less troublesome factor than oil would be if present.

The high cost and large floor space occupied may be regarded as two of the principal objections to the use of surface condensers. With the latter, several plans have been followed in turbine power plants to obtain the maximum kilowatt output for a minimum floor area, among which may be mentioned:

1. The plan of locating the condenser between the foundations of two turbine units.

2. The method of placing condensers alternately at the front and back of adjacent turbine foundations.

3. The plan of locating the condenser immediately below the turbine, and supporting the base of turbine independently on either steel or concrete structures.

Still another method is employed which is desirable when economies in floor space and foundation volume are of first importance, viz., that of making the condenser serve also as the base for the turbine. This combination, although more expensive in some

respects, gives a relatively more compact arrangement than any of the others referred to.

The question whether it is best to install a particular type of condenser in preference to any other depends wholly upon the existing or limiting conditions, and should be solved by the engineer after a careful study of the problem in hand.

Contrary to the general impression, it would seem from Mr. Rockwood's statements in paragraph 13 that the steam turbine is not absolutely proof against injury by water. It is true, however, that nearly all writers have confined themselves to the effect of water carried over from the boilers, and have not given consideration to the possibility of damage from an accumulation of water in the exhaust chamber. Usually the condenser, when of the surface or of the jet types, occupies a position below the exhaust opening of the turbine so that water may drain from the latter by gravity, and thus eliminate the danger which might arise from its accumulation and contact with the rotating blades.

Manifestly some back pressure and consequent impairment of vacuum would be caused by the presence of water in the exhaust pipe when the leakage into the latter from the air seals is at as high a rate as 950 pounds per hour, which is equivalent in amount to more than 10 per cent. of the hourly steam consumption of the turbine at normal full load.

The effect of water in the exhaust pipe when the steam follows an upward course to the condenser has been observed as the result of tests on a 1,500 kilowatt turbine which was mentioned by Mr. Hodgkinson in his paper presented at the June meeting of the Society. In this particular instance the collection of water increased the back pressure from $\frac{3}{8}$-inch to 1 inch of mercury, dependent upon the load conditions.

The subject "Condensers for Steam Turbines" is a very interesting one, and it is to be hoped that the results obtained by Mr. Rockwood at the Atlantic Mills' plant will lead to further investigations and more specific data on this general question.

Mr. F. Meriam Wheeler.—During the last thirty years I have had more or less to do with exploiting different kinds of condensers, including the ejector type, and I must say the latter has many attractive qualities, as for instance, its simplicity and low first cost. It also has its limitations, and also, as Mr. Rockwood truly says, it is often possible to have trouble on account of the liability of water getting into the exhaust pipe, and from thence to the steam

engine or steam turbine. Then it has not the flexibility of the air pump condenser, which latter can be easily regulated to the ever changing conditions of load and ejection water-temperatures. The contracted neck—by means of which the air pump effect is secured—is a fixed part of this style of syphon condenser, and cannot be readily changed. I know we have often had to alter such condensers by substituting larger or smaller neck castings in order to get the best results. With a stationary temperature of water, such as is obtained from wells or mountain streams, and when the engine load is not variable, we get the best results in the ejector syphon condenser. When there is an adequate head of water available, as frequently found where water power has been abandaned for steam, it is a very attractive type of condenser, because it costs nothing for the condensing water supply.

With this type of condenser, however, they have not made the progress as with other forms. I took the matter up, in connection with my business some years ago to try and get over some of the objections, and I brought out a somewhat modified type of ejector condenser to the one named, but I was always longing for a "rubber-neck"—so to speak—to replace the rigid "*Vena contractor.*" If we could only get that contracted orifice in the neck casting, so it would stretch and contract according to requirements, it would be quite an advantage. As I said before, in this so-called "Vena contractor" you get, as it were, a continuous air pump effect in the neck piece.

Many of you will remember the shape of these condensers, so it is not necessary to illustrate what I mean. Another thing that I am a little skeptical about is the statement regarding the vacuum being the same at different points. My experience is to the contrary, in fact I now have in mind a turbine installation across the river where, although the distance is only about 10 feet between the turbine and the condenser, there is a difference in the vacuum of nearly 3-10 of an inch; the exhaust pipe is quite large (30 inches by 16 inches, rectangular shape) for a plant of about 1,000 horse-power, and more liberal than in the case of the Atlantic Mills turbine. I therefore doubt the correctness of the observations, and think it bad practice to use a connection as small as ¾-inch—which I understand was the size of the pipe used for connecting the mercury column to the exhaust pipe and the condenser.

Regarding compactness, it is a very desirable thing to economise floor space, which this ejector condenser accomplishes, but on the other hand, it is necessary to carry the exhaust pipe very high up, which needs considerable length for such piping. The nearer you can get your condenser to the engine or turbine the better, and the tendency nowadays is in that direction. Another thing I have noticed is that this condenser takes more water than the air pump type, as we find by the record of temperatures of the hot-well. There is nothing like having an auxiliary that is adjustable for the varying conditions of load, water temperatures, etc.

Mr. G. R. Henderson.—I would like to ask Mr. Rockwood a question in regard to paragraphs 16 and 17 on page 386. In Paragraph 17 he gives certain estimates on cost amounting to $217 a year which I think represents about 14 pounds of water per horsepower per hour used in the condenser. At that rate for condensing water, the plant would evidently include a cooling tower, and I would like to know whether the cost of the cooling tower is included in that estimate.

Mr. H. H. Suplee.—There are one or two points in the paper which I think might be mentioned in connection with Mr. Rockwood's response. In the first place the vacuum is given as 28 to 28½ inches, but nothing is said about atmospheric pressure. Mr. Rockwood referred to that for a moment in his remarks, but considered it of comparatively small effect, I believe. As a matter of fact it has varied at least two inches in the course of the past season, running as low as 28½ and as high as 30½, so there is room for variation. In matters of this sort it seems to me that the atmospheric pressure ought always to be given if possible with any statement of the condenser pressure, because the one depends so directly on the other that it is almost impossible to compare the results, unless we know the height of the barometer. Every station ought always to have a good barometer.

In regard to the question of the ejector condenser being better than the surface condenser, it is very interesting to note that Mr. Parsons has in his recent work in England called upon the ejector condenser to help out the surface condenser. I have here, and I will be glad to incorporate it in the printed discussion an excerpt from a paper by Mr. Parsons. The principle is simply this—that an ejector is used to draw the air from the main surface condenser, live steam being used. This jet delivers into a smaller

auxiliary surface condenser, the water passing through both of them, and the air being drawn from both of them, the main condenser being separated from the smaller one by a water seal. The result is that the air is drawn from the main condenser and delivered to the air pump with the production of a vacuum of from 1½ to 2 inches better in the main condenser than there is in the air pump. The jet draws the air into the air pump and enables the vacuum to be improved by nearly two inches. Against this is to be charged the live steam required to do it. Mr. Parsons states that the efficiency of the surface condenser is improved by the removal of the air, because the air forms a film on the outside

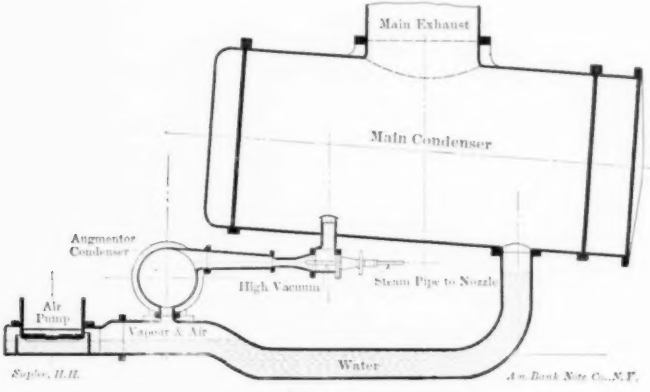


FIG. 135.

of the condenser tubes reducing their conductive power and very greatly lowering the capacity of the surface condenser.

As shown in the illustration (Fig. 135), the augmentor jet draws the air from the main condenser, delivering to an auxiliary condenser, the latter being about 1-20 the capacity of the main condenser. The jet sucks the residual air and vapor from the main condenser and delivers it to the air pump. The jet consumes about 1½ per cent. of the steam used by the turbine at full load, and this steam, which would otherwise impair the vacuum, is condensed by the auxiliary condenser. This device has been applied with success to the turbine steamer Manxman, and is described very fully in a paper presented before the Institution of Electrical Engineers by Mr. Parsons last May.

The President.—Will you tell us what type of air pump is used there?

Mr. Suplee.—It is a vertical air pump, leading from the lower main and drawing the water from both condensers. I think it is the Edwards type.

Mr. George J. Foran.—I would like to say first in reference to Mr. Suplee's remarks, that I hardly think if the non-condensible vapors were properly removed from the condenser, there would be any increase in the vacuum.

I have understood that it was only advertised or claimed that the so-called "Augmentor" would increase the vacuum with the ordinary type of condenser.

We have made a special study of turbine condensers, and the design is such that there can be no air pockets or body of non-condensible vapor remaining in the condenser, and we have been able, for example, at St. Louis in connection with the General Electric turbine, to maintain practically theoretical results as to vacuum and temperature, which can hardly be improved upon.

While it may be possible to decrease the size of the vacuum pump by the attachment of the Augmentor, I do not see how it can increase the rate of heat transference of the condensing surface in a condenser which is properly designed for the withdrawal of the non-condensible vapors, as the temperatures of the condensing water and vacuum would remain unchanged.

Although I have seen a number of papers and reports upon this apparatus, I have never yet seen any definite statement as to the amount of surface, temperature, weight of steam or other data necessary to check up the actual performance. I am sure there are a great many of us who are interested in the matter who would be glad to have information as to actual results obtained.

With regard to Mr. Rockwood's paper I would like to say that, in common with all the members of the Society, I am glad to hear another of Mr. Rockwood's papers. They are always interesting, and we know that Mr. Rockwood is a keen observer and reports results just as they appear to him without favor and without regard to precedent. I have been asked by so many members to discuss this paper in some way that I finally decided to do so, although I believe as a general rule it would be better in a case of this kind for the manufacturer of a special line of apparatus under discussion to refrain from active participation rather than to run the risk of precipitating a purely technical and possibly acrimonious discussion between builders of competitive apparatus,

which would certainly be out of place in a scientific meeting of this character.

The company with which I am connected manufactures every type of condensing apparatus referred to in the paper, and my remarks are as applicable to machinery of our own as any other of the type specified. The manufacturer of condensing apparatus is obliged to produce the type of machinery demanded by the engineers and users. It so happens that there are hardly any two turbine units which are quite the same in all details. Each engineer has to decide which type of condenser best suits his conditions—as a rule the surface condenser appears to have been preferred by both the turbine builders and engineers—for three principal reasons.

Saving of feed water.

Vacuum obtained practically in the turbine.

Shorter pipes with no frictional loss and less chance for leakage.

The type of auxiliaries is a special study in each case—as a rule independent units are preferable, and they should be steam rather than electric driven. The proper selection of type depends upon many other conditions—the nature of existing units in the plant—the use of economizer, heaters and oftentimes the special conditions of load and service.

All the condenser man is asked to do usually is to "sharpen his pencil."

I believe Mr. Rockwood's paper was presented with the idea of calling attention to the use of a different type of condenser than that ordinarily used in turbine work—with the possible benefits following a general discussion. A careful discussion of the plant referred to is not possible, as no drawings or definite data are furnished upon which to base actual comparisons.

I do not think the ejector costs given include enough apparatus to make them a fair basis of comparison. I would like to call attention to the statement in section two with reference to air pumps. There are many who consider the single stage pump preferable.

In an injector or ejector condenser the direction and velocity of the condensing water is preserved and the velocity increased by the condensing steam (as in any injector) to a velocity sufficient to carry away the non-condensible vapors through a suitable throat—and its efficiency depends upon a reasonably constant ratio of the steam and condensing water. In the so-called elevated

condenser this principle is not utilized. Steam is condensed in the regular type of jet condenser and, with the condensing water, falls into and passes out by gravity through a 34-foot tail pipe against the atmospheric pressure, carrying along some of the non-condensible vapors, most of which, however, are removed by a positive displacement vacuum pump. But in both these condensers as ordinarily installed, the circulating pump discharges directly into the condenser, its work being reduced by the assistance of the vacuum to an extent of about 25 feet in practice. In the Atlantic Mills Mr. Rockwood has thrown away this 25-feet assistance and delivers the water to the full height of the condenser.

He does not state whether this was solely to free the water of air as alluded to in section 7 or to enable the water to enter the condenser at a higher velocity, thus passing through the throat at a higher velocity and producing a more satisfactory "air pump," as he terms it in section 9.

Whatever the purpose, the work has been much increased, and it would be far more economical of power to use the ordinary elevated condenser with its extra vacuum pump, gaining at the same time in the range of efficient capacity of the condenser.

Based upon usual conditions the condensing water for 400 kilowatts would be approximately 1,200 gal. per min. and the head at Atlantic Mills is apparently about 31 feet. This amounts to a little less than 9½ horse power of actual work performed. Under the usual conditions of delivery to an elevated condenser carrying 28-inch vacuum the head would have been about five feet or an actual work of less than 2 horse power, and the actual work of the vacuum pump on this elevated condenser would be less than 3 horse power, showing a net saving of approximately 4½ horse power, or nearly 50 per cent. over the injector condenser. Furthermore, the vacuum pump would be fully as efficient a machine as the centrifugal for power drive.

Even if the conditions assumed are not correct and the quantity of water assumed incorrect, the amount of vapors will also vary and the difference in work would still be proportionately large. I note that the 6-inch pump is rated by the builder at 1,200 gal. per min. capacity and the shaft horse-power given as 0.65 per foot of lift.

I have already stated my belief as to the reasons why Mr. Rockwood has presented his paper, and I do not understand that he

claims the ejector type is superior to the elevated condenser for their work. It certainly is not if it becomes necessary to increase the circulating work to the extent shown.

As a matter of fact, under certain conditions it is possible to operate the elevated condenser at high vacuum without the use of a vacuum pump. We have had a number of such condensers operating for some time and carrying vacuums varying from 27½ to 28½ inches with the vacuum pump idle—and quite likely condensers of other makes have accomplished similar results—and this without increasing the head upon the circulating pump.

It is true these condensers are not attached to turbines, but so far as I know no one has ever told the condensers that yet.

Mr. Wm. E. Crane.—It is hard to argue against facts, and I think that Mr. Rockwood is able to substantiate all that he has said. I speak from my own experience which has extended over twenty years with this type of condenser, as well as with a number of other types, and my experience has not been sitting in an office and taking reports which have come to me from the observations of others, but it is what I have actually seen in my own practice. It was my practice to always put a gauge at the nozzle of the exhaust in the cylinder, and I have had no trouble with this type of condenser in maintaining with a load 28 to 28½ inches of vacuum with a mercury gauge. Of course, when you have a light load the vacuum will fall off a little. It has been mentioned that the surface condenser was superior to produce a vacuum. Now it does not take but a very small amount of air to knock down a whole lot of vacuum. All you have to do is to give it room. Where you have a surface condenser you have room for this air to expand, but with this type of condenser under discussion there is little room, and the air is down the tail pipe before it has a chance to do mischief.

The remark has been made that you have to pump water into this condenser. Which costs the most—pumping the water where the vacuum is helping you or pumping against a vacuum?

*Mr. G. I. Rockwood.**—I will try to answer the points that have been raised, though I may not manage to do them all full justice within the brief time at my disposal.

Dealing first with the discussion by Mr. Hodgkinson; he says he is surprised that the ejector condenser shows so high a vacuum

* Author's Closure under the Rules.

without the aid of a separate air-pump. I cannot think of any reason why he should be, as I called special attention, in his presence, at the Saratoga meeting, to the high vacuum there recorded by Professor Jacobus in the test of a steam engine of about the same power and using the same size and type of condenser.*

Moreover, and as I also pointed out in that discussion, there is the additional reason why a turbine should be expected to operate more continuously without air leakage under a high vacuum than a reciprocating engine might do, namely, the absence in the design of the turbine of anything corresponding to the piston-rod and valve-stem packings of the ordinary engine. These packings introduce more or less of an opportunity for air-leakage to such an extent as to seriously affect the steadiness of the high vacua required in turbines for the best results; and therefore, it is fortunate that when once mechanical tightness against air-leakage has been secured in a turbine exhaust system arranged as described in the paper, all further anxiety as to maintaining the original high vacuum is at an end.

This also answers what Mr. Hodgkinson says with regard to the inadvisability of providing an air pump as an insurance against further leakage. In a word, if a good mechanical job has been done in the first place, there will be no further leakage.

It is sometimes the case that air is not entirely excluded from the injection water. This may occur where direct pumping from pond or river, or most likely tide-water, without the intervention of a tank, is resorted to. The vacuum gauge will vary between wide limits under such circumstances unless the suction pipe is sealed by water its entire length, and either an air pump or a tank becomes a necessity; but a tank will always be the better cure.

An air pump certainly cannot be operated for nothing. In the first place, if you are going to operate an air pump, you have got to buy it; then you must have a place in which to put it; a foundation to put it on; an engineer to oil, clean, repair and operate it. The steam that passes through the driving cylinder may not be a great quantity, but this is the least expense connected with owning an air pump—just as the expense for gasolene is the least connected with the expensive pleasure of operating an automobile.

Mr. Hodgkinson doubts the correctness of my tests to determine the drop in pressure between the turbine and the condenser. I

* See page 1283, Vol. XXIV.

will reply that, except for one influence which may have an effect upon the reading of the mercury column, I think those tests were accurate within limits so narrow as to be of no technical importance. It may be that the end of the pipe screwed into the turbine casing was affected by a Pitot tube action which was different in its intensity from the similar action upon the end of the pipe where screwed into the elbow on top of the condenser; but I am confident that except for an error possibly introduced in that way, the reading of the mercury column truly represented the pressures in the turbine exhaust chamber and in the elbow on the condenser, respectively.

I have lately visited several turbine power stations, partly with a view to observing what actual indications of their different vacuum gauges were to be seen. In one plant, provided with Edwards air pumps and a magnificent surface condenser installation quite capable of attaining under favorable conditions as much as 29 inches or over, the actual fact was 27 inches. In another case, where the turbine had been in operation but a few months, the vacuum was 26 inches. In other words, not all the air leaks had as yet been discovered. In another, which had been running but a few days, the vacuum was but 22 inches. So the practical advantage of having comparatively so little chance for air leaks in the ejector condenser system, as compared with the large and difficult joints to render tight in the first place, and afterwards, to keep tight, that are so numerous in the surface condenser system, this practical advantage, I say, is of considerable importance.

Mr. Joseph Morgan's valuable discussion presents numerous facts for consideration. He doubts the ability of the ejector condenser to maintain the vacuum under wide variations of load, and advances the theory that considerable pipe-volume in the exhaust pipe system is a great aid under these circumstances.

His experience with variable loads is greater than mine; and yet, I see nothing more in his statement of his experience than that he either did not get his condenser big enough to condense the maximum amount of steam he actually had to provide against, or else his exhaust system was not tight, and a large increase in the steam and water supply meant the introduction of more air than the size of throat was adapted to remove. This would be very likely to be the case with rolling mill engines, because they start off with great rapidity at full stroke cut-off and temporarily

discharge large amounts of steam compared with their discharge directly the work is finished.

Mr. Morgan gives a very good illustration of the truth of the opening paragraph of the paper, even in the case of the jet condenser and air-pump arrangement, in his drawings submitted of the Johnstown, Pa., turbine plant; although my remarks apply more especially to the surface condenser type of equipment. With the Weiss condenser of this plant wholly on the outside of the building, the turbine room is given up, to the extent of one-half of its entire area, to the air and circulation pumps and piping.

Mr. Henderson asks to know whether the "cost of the cooling tower" has been included in my estimates. It is not included in any case, as its cost is a factor which does not affect the solution of the problem as to what type of equipment it is best to choose.

The barometer readings asked for by Mr. Suplee were absent from the paper because they were not taken; and they were not taken because there was not time to complete the data for this plant in season for presentation at this meeting of the society.

Mr. Perry gives a clear statement of the "state of the art" of condenser practice.

Mr. Foran is in error in some of his comments, but through no fault of his own, as there is a regrettable misprint in the paper of *inches* for *feet*. Thus in Paragraph 6, third line from last, it says that the level of the water in the tank is maintained "six inches" below the water inlet nozzle, where it should say "six feet." *

This misprint affects the accuracy of Mr. Foran's figures twenty-five per cent. I may say, however, that there were certain advantages incident to the operation of this turbine in having the tank elevated as much as described.

First was the uncertainty whether the condenser would draw sufficient water, under all conditions of load and temperature, to maintain 28 inches vacuum if the tank were placed any lower. We have an answer to this now in our experience, and if I had it to do in another case I should put the tank at an elevation eight feet lower still, to reduce the power required to lift the water into it.

Second: With the tank at that particular elevation described,

* EDITOR.—This error has been corrected.

the water supply for sealing the spindle at its exit from the turbine casing was of exactly the right pressure (about 4 pounds per square inch), and of invariable amount and pressure, as it could then be taken from the tank.

Third: The tank, at its actual elevation, was cheaper to install and operate, as it stands directly on the turbine floor and in the turbine room.

If the tank were lowered eight feet below its actual level, Mr. Foran's figures for cost of pumping injection water must be cut in two. But furthermore, Mr. Foran selects too great a volume of cooling water as requisite for the injector type of condenser. It need not be over three-quarters the volume needed in a surface condenser system, because the water mingles directly with the steam instead of being separated from it by condenser tubes.

The calculation should be, for this case with the tank lowered 8 feet, as follows:

400 kilowatts = 625 indicated horse power.

$625 \times 13 = 8{,}125$ pounds steam condensed per hour.

$8{,}125 \times 38 = 308{,}750$ pounds water raised 20 feet per hour.

$$\frac{308{,}750 \times 20}{33{,}000 \times 60} = 3.$$ indicated horse power theoretical, or about six horse power at the belt of the centrifugal pump.

This is about one per cent. of the indicated horse power, and is a more economical performance than anything ever done by an air pump equipment. It may be compared with the performance of the Johnstown condenser plant referred to by Mr. Morgan. This plant takes 12 horse power for the same load, i.e., 400 kilowatts as against 6 horse power in the case of the Bulkley condenser.

In further reply to Mr. Foran I may say that, for reasons of economy of space, regularity in maintaining the vacuum, cheapness in first cost and, finally, because of their undoubtedly greater economy of operation, I do claim that, for *steam turbine* work, the ejector condenser is superior to any other type whatsoever except, as stated in the paper, where the surface condenser must be employed. I understand, however, that it is being experienced in the New York stations that electrolysis is an enemy to the tubes of surface condensers of considerable importance, even to the extent of causing the condensed steam to become brackish and hence unfit for feedwater.

No. 1073.*

DIAMOND TOOLS.

BY GUS C. HENNING, NEW YORK, N. Y.
(Member of the Society.)

1. As nothing has yet appeared in the *Transactions* of this Society relating to the above subject, and which must be of interest to engineers and manufacturers, the writer proposes to present a short description of diamond tools.

2. Steel is, of course, the one material in almost universal use for cutting and working stone, metal, wood and other materials, because of its great strength and the degree to which it can be hardened. There are some materials, however, which, because of their hardness, structure or non-conductivity of heat, cannot be worked economically by means of steel tools. The latter become worn rapidly, losing their shape and dimensions to such degree and extent that the work produced becomes inaccurate, causing constant interruption of operation, loss of time, and the use of new tools or frequent regrinding or shaping of the old ones. This causes great expense and delay in production.

3. The great friction produced by cutting materials in some cases draws the temper of steel tools, making them useless.

4. Hard rubber, paper and hardened steel cannot be readily worked by use of steel tools, as is also the case with hard stone. In these cases a much harder material is required, and for this reason diamond is used. The diamond which is used is of two kinds, totally different in appearance and quality.

Black Diamond.

5. This has a very dark purple brown color, is an amorphous, granular stone with rarely any crystallization visible or traceable,

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

and is called Carbon or Black Diamond. It is the hardest material known and has great strength.

Bort.

This, called Bort, is entirely crystalline, and generally transparent and of all colors of the rainbow, as well as clear and transparent as glass. The latter is considered of greater hardness than all other bort except some which is almost black. Bort is extremely brittle and is readily fractured or "cleaved" in the three directions of its cleavage planes parallel to the sides of the octahedral crystal, in which shape it is most commonly found. The dodecahedral crystals are also readily cleaved in a similar manner.

7. In spite of the very great hardness of all kinds of diamonds, they are readily sawn, drilled, cut and polished; carbon (black diamond) cannot, however, be polished, as is the case with bort.

8. Diamond cuts diamond, while steel saws and drills and cast iron discs, charged with diamond dust, are used for the other operations.

9. All kinds of grinding wheels, being made of extremely hard materials, are most readily kept free from filling or glazing and in perfect shape by diamond tools.

10. In certain classes of work, where great accuracy and precision are primary requirements, or extremely fine lines are essential, the diamond is the only material that answers the purpose. Thus lithographers, engravers and scale-makers use them for fine work.

11. There is another very important field of production in which diamond is all but imperative to obtain satisfactory results at reasonable cost, viz., that of wire drawing.

12. Formerly all small wire was drawn through holes in hardened steel plates, but these wear so rapidly that the wire soon loses its calibre and becomes unround. As it is all-important, especially in electric work, that the wire be of absolutely uniform size, so as to maintain constant resistance and permit symmetrical distribution of weight about spindles and shafts, it became necessary to use a material harder than steel, and hence diamond was again resorted to. This made it possible to avoid delays in replacing worn dies, and because of the great permanence of accuracy of the calibres of the holes in the diamonds, materially re-

duced the cost of producing fine wire of copper, brass, steel, iron, nickel, and of other metals.

13. It is of course well known that stone is drilled and sawed

FIG. 136.

by the use of diamonds, these having been used in core drills, which, in an extreme case, have cut solid cores of about 21 inches diameter.

14. In diamond drills, stone saws and grinding wheel dressers,

FIG. 137.

the rough diamond is used in appropriate holders, set either by staking, brazing, soldering, or by casting molten steel around the diamonds.

15. A peculiar property of the diamond is that it can be plated

FIG. 138.

like any metal: this property is made use of in the galvanoplastic setting. The galvanoplastic setting consists in first plating the diamonds and then casting other molten metal around them, which

alloys with the deposited metal. Thus an absolutely firm and rigid setting is produced.

16. Very high temperature does not affect the diamonds either in their hardness nor, when sound, in their solidity, and does not produce checks or other flaws. A temperature higher than that sufficient to melt steel will, however, burn the diamond, and that of the electric arc will do so readily.

17. The diamonds in tools used for doing accurate work are,

FIG. 139.

however, all "shaped" by cutting and polishing, so as to imitate the customary shapes of steel tools.

18. Glass and china are also drilled by shaped diamonds, in which case a triangular splint is generally provided with a flat triangular pyramidal point, as shown at Fig. 136, which, when using turpentine as a lubricant, penetrates glass and china more readily than any other tool, and lasts for from one to two years, unless broken by carelessness or accident.

19. Fig. 136 shows a round-nose tool. The diameter or curve of cutting edge and its clearance from a right angle can be produced at will.

20. Fig. 137.—This shows a sharp-nose tool, and in this again the angles can be varied at will.

21. Fig. 138—This shows a square-nose tool, which is used mainly for producing very high finish of hard rubber.

22. Fig. 139.—This shows a tool commonly used for turning paper calendar rolls, which must be very accurate and smooth. A piece of black diamond shaped as a round-nosed tool is simply clamped in proper position between two pieces of steel.

23. Tools shown at Figs. 137, 138 and 139 are most commonly used for turning hard rubber, because the diamond lasts a very

long time without wear and produces absolutely smooth and accurate work, while steel tools wear off in a few minutes at most and get very hot.

24. One other reason why it is economically advantageous to use diamond tools for turning hard rubber is that very high speeds can be used, 450 to 500 feet per minute being common.

It may here be added that diamond tools are most suitable for working carbon used for electrical purposes.

25. Fig. 140 shows a diamond die for drawing wire.* This is a bronze block in which is set a diamond perforated by a tapering, polished hole through which the wire is drawn. The holes in these diamonds are rarely made larger than 0.064 inches diameter, because steel draw plates or dies are considered sufficiently accurate

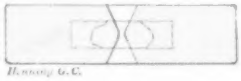


FIG. 140.

and economical for larger sizes, and because of the great cost of diamond dies. The smallest dies which have come to the notice of the writer had holes of 0.001 in. diameter, although clients have called the calibers as small as 0.00045, 0.00055 and 0.00065. It is common practice to make the holes in diamond dies accurate to a 0.0001 inch, which to many engineers may seem almost impossible, and is therefore here mentioned.

26. In drawing copper wire, it is customary to draw a 0.064 wire in one pass from a rough wire of 0.072 diameter. Smaller sizes are then produced by the following consecutive reductions: to 0.053, 0.045, 0.040, 0.036, 0.032, 0.028, 0.025, 0.022, 0.019, then by 1-1000 down to 0.0075 and by half-thousandths down to 0.001.

27. It may be mentioned that diamonds wear increasingly when drawing the following metals in the order stated, viz.: gold, silver, copper, brass, bronze, platinum, soft steel, nickel, iron and crucible steel (piano wire).

28. In order to show why such expensive material as diamond can be used economically it may be stated that diamond dies wear

* This wire die is shown in section in order that its construction may be clear; in the most improved setting steel is cast around the diamond to prevent its bursting, and bronze is then cast around the steel setting to prevent corrosion of the steel by the substances used to lubricate the wire as it passes through the die.

up to eight years under constant use. One die of 0.004 caliber has, according to the record, drawn over 550,000 pounds of soft copper wire.

29. Diamond drills for drilling glass wear from one and one-half to two years before requiring recutting.

30. As is well known, diamonds are also used for spindle bearings in watches, and most recently have been introduced as cupped

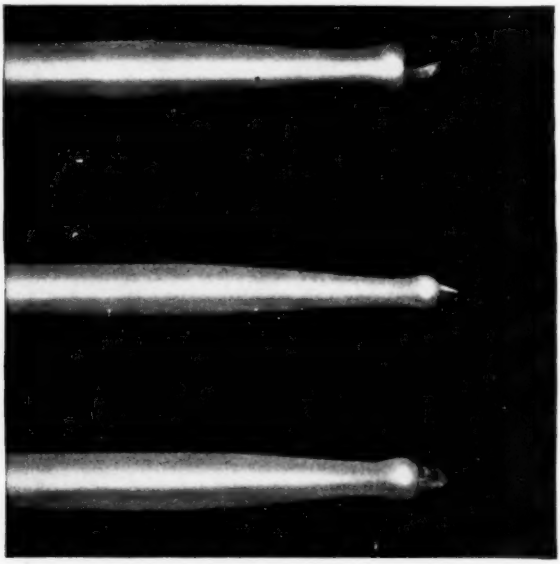


FIG. 141.—DENTAL DRILLS.

bearings for the pivots of electric meters, because they produce the minimum of friction, and do not wear out in many years.

31. Another purpose for which diamonds are used is that of drilling teeth, especially artificial teeth. In these drills minute chips of diamonds soldered into steel shanks are used. These diamonds are not prepared in any manner, as their points and edges when properly selected are sufficiently hard and sharp to penetrate bone and porcelain.

32. The shapes of chips most generally used are flat, triangular points and three-sided pyramids. The most perfect drills for this purpose have diamonds of triangular sections with a pyramidal polished point.

These drills are illustrated on a 6-fold scale at Fig. 141.

DISCUSSION.

Mr. Fred. W. Taylor.—I would like to ask Mr. Henning what shape he considers best for turning emery wheels. I fail to find this mentioned in his paper.

Mr. Gus C. Henning.—The shape of the diamond has little to do with the dressing of wheels, but if it is desired to produce some particular shape, a sharp point, round or square edge is selected. The general rule is, *the larger the wheel, the larger the diamond to be used.* Only rough stones are used for dressing grinding wheels. Grooves are cut by use of pointed stones.

Diamonds are sometimes run under such pressures by unknowing or careless users that they become red hot, in which case they are ground away very rapidly. It is imperative to use them under a flood of water to keep the temperature down, and insure good work and long service.

Mr. Taylor.—In truing up emery wheels everyone has had the experience of losing the diamonds. The opinion I have formed is that the diamond becomes loose and is lost, due to the action and the heat resulting from too heavy feeding. I would like to have Mr. Henning's opinion.

Mr. Henning.—With proper methods and careful workmanship it should be a simple matter to set a diamond so that it will neither flake, break nor become loose. The large end of the diamond should be deeply set in steel, and unless it is broken by excessive pressure it will never come out, but will remain firmly set until completely worn down.

The best method for setting the diamond is to cast steel around it, and if the proper casting temperature is used the diamond becomes practically cemented in place. The proper temperature is about 4,500 degrees.

When an octahedral or egg-shaped diamond is used it may be ground down to its maximum section; this being reached, the diamond should be reset end for end to expose the back end. This done, the diamond can again be used until its middle section is reached. This leaves a thin slab of diamond in the setting, which can again be reset. When octahedral crystals are used it is necessary to reset them four times, once for each point. This will ensure that the entire diamond can be used up.

Every diamond should be carefully catalogued in a record as to its provenance, weight, shape, dimensions and general char-

acteristics. By such record its particular field of usefulness can be designated.

Prof. W. W. Bird.—I should like to have Mr. Henning give some information as to the hardness of the diamond.

*Mr. Henning.**—There is very little known about the hardness of diamonds, or about their relative hardness, except that they are the hardest material known. We know by rough approximation and guessing that some kinds are harder than others, but correct or accurate investigations have not been made as far as the writer has been able to learn.

Carbon (or black diamond) found in Brazil is entirely granular and amorphous, and has no distinct cleavage planes in three directions and is not crystaline; it is opaque, and while having a pinkish-gray fracture, is generally purplish-gray or brown on the surface. This is supposed to be the hardest variety.

Crystaline diamond (bort) has three distinct cleavage planes, but is found in all colors of the rainbow, is transparent, and when absolutely clear is used for jewelry. The latter is considered very hard, but not as hard as a variety which is partly black but translucent. The hardness of others is supposed to vary in accordance with the colors in following order: Light brown, yellow and dark brown. It has repeatedly happened that the latter explode upon removal from the mine and exposure to sun, light and air. Brazilian and Australian diamonds are generally considered harder than African stones.

The hardness of diamonds is commonly tested by application to a grinding wheel and noting results. This is, of course, but a mere guess.

A few words in regard to diamond dies. The wires drawn through them for electrical purposes must be so accurate that in winding coils, for example, the resulting coils should be perfectly symmetrical. For electrical measuring instruments wire of extreme thinness is sometimes used, and all of it must be of greatest accuracy to produce correct resistances. Also platinum wires for use in incandescent lamps must be perfectly round and smooth. The dies used must therefore be perfect. The holes are made by drilling and polishing, and very small variations in size are often required. Provided the copper is pure, there is very little wear on these dies and most of them are destroyed by bursting.

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* Author's closure under the Rules.

Probably the largest diamond dies made have been 0.74, 0.64 and 0.53, but very few of these are used. Ordinary sizes, from 0.45 to 0.30, are made in quantities.

I also wish to call attention to the behavior of the diamond as a cutting tool. It is so hard that when shaped and polished to a cutting edge it will not wear except after long use. Its destruction is almost invariably the result of careless handling which results in its striking or being struck a hard and sudden blow.

When used for cutting rubber the diamond will sometimes chip, but this is because it has struck some piece of foreign matter such as metal, which has been mixed in with the rubber. Ordinarily it would cut this metal, but the speed of cutting for rubber is much too high for metal cutting and hence the diamond is chipped.

The diamond tool is used not only for finishing but for roughing as well, and leaves a very fine finish which only requires polishing.

For truing metals or stone the diamond is not used because of its cutting ability, but because of its hardness and the consequent wearing away of the softer material.

Hardness in the case of the diamond means not an ability to resist indentation or crushing but an ability to resist abrasion. It will not wear, and therefore holds its shape. For this reason even-tempered steel can be cut with the certainty of obtaining accurate results.

No. 1074.*

MORE EXACT METHODS FOR DETERMINING THE EFFICIENCY OF STEAM GENERATING APPARATUS.

BY A. BEMENT, CHICAGO, ILL.

(Member of the Society.)

1. The accepted method of conducting tests of steam-generating apparatus determines the results of the performance of the combined apparatus and its manipulation. Such experiments are referred to as "boiler" tests, yet they are not properly tests of the boiler, nor do such experiments show the results of the performance of the fire or the furnace feature, except as measured through a more or less inefficient boiler. Basing results on the coal supplied to the grate less that which goes to the ash pit or is removed from the fire in cleaning, is a step toward determining the efficiency of the boiler, but, with present methods, it is assumed that the effect of the fire is constant or uniform, although gas analysis is required which shows that fires differ in quality, but this fact is neglected in considering the efficiency of the boiler, yet, any performance is dependent on the condition of combustion, or, in other words, the temperature of the fire.

2. The purpose of the boiler is to receive heat from the gases, which is supplied to them by the process of combustion. As heat only flows from a region of higher to one of lower temperature, and as its rate of flow is governed by the difference between the temperatures, it necessarily follows that most heat will flow to the boiler when its temperature is highest. It is useful in this connection to consider that heat and temperature are two entirely different things, temperature being the condition under which the heat is present. If a given quantity of heat is contained in a large amount of gas, it will be at a low temperature, but if in a small

* Presented at the New York meeting (December, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXVI of the *Transactions*.

amount, it will be present at high temperature. That this is true may be determined according to the formula,

$$\text{Temperature} = \frac{\text{Heat developed}}{\text{Weight of gas} \times \text{its specific heat}}.$$

3. Thus the performance of a boiler is influenced by the temperature at which the heat is supplied; also, when a large quantity of heat is furnished, its absorption is less complete than with a small quantity, the temperature being the same in each case.

4. Therefore, if boilers are to be compared with each other, so their heat efficiency can be determined, it could be accomplished if supplied with the same relative quantity of heat at the same temperature. These conditions would be fulfilled if a uniform number of heat units be supplied for each square foot of heating surface per unit of time and at some accepted standard temperature.

5. Gauging the required quantity of heat to be supplied would present several difficulties. The heating power of the fuel could be determined by sampling an analysis previous to instead of after the test, which would enable the proper amount of heat in the coal to be furnished to the grate, but fuel lost in the ash-pit, removed in cleaning the fire or carried over the bridge wall, and incomplete combustion, would have a varying influence of the quantity of heat generated, and the percentage of such heat, undeveloped, must be known and provided for in the supply of the fuel to the grate. The use of a standard high-grade coal would, of course, very largely reduce these difficulties.

6. As different steam pressures have an influence on the amount of heat flowing to a boiler, it would be necessary to use some standard pressure, which need not be that which would be employed in service. And the loss of heat by radiation would be determined or not, as circumstances require, but with well-set apparatus it could be assumed to be constant at rated capacity.

7. The supply of a certain predetermined amount of heat would be a very difficult matter, and a less efficient boiler would absorb a smaller quantity of heat than a more efficient one, although the same quantity be supplied per unit of surface and time, and at the same temperature, therefore, the less efficient boiler would do less work. Thus the capacity would not be the same for each, and instead of there being simply a difference in efficiency there would be a difference in both efficiency and capacity. It would

probably be better to produce a uniform capacity as measured in equivalent pounds of water evaporated from and at 212 degrees per square foot of heating surface per hour, or, in other words, operate the boiler at its rated horse-power capacity, for example. This would be a simpler method than that of affecting uniform supply of heat, and a standard condition of combustion as shown by the gas analysis would insure that the heat be developed at the required temperature. Owing, however, to slight varying composition of fuels, it would be desirable to use coals in which the ratio between carbon and hydrogen is approximately the same. Thus with standard conditions of developed capacity, initial temperature and steam pressure, the heat developed referred to the heat absorbed, will show the efficiency of the boiler, so that comparisons may be made.

8. The efficiency of any particular boiler is a fixed quantity, which once ascertained according to some standard applies to any number of reproductions, and thus need only be determined once. It does not follow, however, that the efficiency of different sizes of the same kind of boiler is the same. For example, there is a horizontal water tube type which is usually made from 9 to 14 tubes in height, the travel of the gases being three times across the tubes. The higher boilers are more efficient than the lower ones. For the reason that the gases pass across the tubes at a volume dependent on their temperature, and as the temperature drops, the volume becomes less, which results in a portion of the tube surface of the second and third passes especially not being in the path of the flowing gases, therefore the fixed path of the gases passes across 27 tubes in the boiler which is 9 tubes high, and across 42 tubes in the one which is 14 tubes high.

9. Purchasers' requirements will demand in the future as in the past that tests of the combined apparatus with the fuel to be used be made to demonstrate if a certain efficiency and capacity are obtainable. The results of such tests, however, are confusing when an attempt is made to determine the efficiency of boilers or of any portion of the apparatus.

10. It is the author's opinion that there is a greater difference in the efficiency of different boilers than is generally realized. In this connection, experiments on two boilers to determine their relative efficiency may be of value.

11. These boilers of a horizontal water tube type adjoined each other in one battery. With each the gases were discharged to the

same flue and steam at 175 pounds pressure to the same steam header. A chain grate stoker was located under each, and the furnace feature of the apparatus was well developed by fire brick tile suspended from the bottom tubes of the boiler, which extends approximately 12 feet back from the front, corresponding to a brick arch of that length, in addition to the ignition arch of the stoker. These boilers are designated as *A* and *B*, and they and their associated features are exactly the same, except that of the path of travel of the gases among the tubes. With *A* the gases travel with the length of the tubes once and do not act on all of the tube surface. With *B* the travel is three times the length of the tubes, in separate passages, which caused the gases to come in contact with very much more of the tube surface than with the boiler *A*. Table No. 1 gives the relative final temperatures of escaping

TABLE NO. 1.

	Boilers. A.	B.
Condition of combustion, CO_2	10.7	10.3
Horse-power developed	564.3	580.4
Heating surface, square feet	4,800	4,800
Temperature of escaping gases, final	657	469
" " " " above air supply	582	394
" " " " " steam temperature	278	92
" " " " greater than B	186	....
" " steam	379	377

gases at approximately the same capacity and condition of combustion. These experiments were not made at the same time, but the same kind of coal was used. The reason that the gas analysis was only carried to CO_2 is that, owing to the very efficient tile roof furnace, combustion was always complete. The horse-power developed placed each boiler nearly on the same basis, being about 3 per cent. to the disadvantage of *B*, therefore the work done by each was approximately the same. The condition of combustion was also slightly in favor of boiler *A*, but this may be considered as approximately the same for each. Thus all conditions were practically the same except that of temperature of escaping gases, which showed a considerable difference in the boilers as gas coolers.

12. If the specific heat of gases at high temperature were known, the gas analysis and an ultimate analysis of the fuel would afford data for calculating the quantity of heat in a pound of gas at initial temperature, and the heat remaining at final temperature

could be calculated from direct temperature measurement, and the efficiency could be expressed according to the equation,

$$E = \frac{H_1 - H^2}{H_1},$$

and it would not be necessary to determine the quantity of fuel used in such an experiment as given in Table No. 1.

13. If the reduction in temperature of escaping gases from a boiler by 186 degrees may be taken as representing the work of an average economizer, then it appears boiler *B* possesses an efficiency equal to that of the combined efficiency of boiler *A* and an economizer; or, in other words, that *B* is as efficient without an economizer as the combined efficiency of boiler *A* with one, which serves to illustrate the great difference which may exist in different boilers.

14. These boilers were also tested by the alternate method of the standard code, which was feasible, because conditions could be exactly the same, and the object was to compare one with the other, and not with boilers in general. The tests were run at the same time, one being started 15 minutes before the other. Alternate wheelbarrows of coal from the same source were laid down before each boiler only as required. The thickness of the fire was the same. Each damper was full open and the speed of the stoker regulated so that the fuel bed extended to the end of the grate. The result is shown in Table No. 2.

TABLE NO. 2.

	Boilers. A.	B.
Pounds of pure coal burned per hour	1,765.75	1,446.56
" " water evaporated per hour	13,210.34	13,629.24
" " " " per pound of pure coal burned	7.47	9.42
Heating power of pure coal B.t.u. per pound	13,633	13,633
Temperature of feed water	212	212
" " gases, final	594	442
" " " above air supply	514	362
" " " " steam temperature	217	65
" " " greater than B	152	....
" " saturated steam	377	377
Draft over fire, inches of water	0.38	0.22
Horse-power	383	395
Efficiency, per cent. of heat absorbed by boiler	52.93	66.73
Fuel saving of boiler B over A, per cent	.....	20.68

15. Pure coal is an expression which the author prefers to use instead of that of combustible, because the fuel free from ash and

moisture contains oxygen and nitrogen, which are not combustibles. The capacities were lower than those in Table No. 1, because the coal used was a less favorable size. This latter test shows the effect of the difference in efficiency of the two boilers in ordinary service.

16. The author prefers to recognize a steam generating apparatus as being composed of separate features as follows: As boiler and grate, with this combination, the gases flow in contact with the heating surface immediately as they leave the fire. Also, as boiler, grate and furnace, the latter feature consisting of a refractory roof over the fire which extends back a considerable distance, its presence making it possible for the gases to be mixed together a sufficient extent while at high temperature, so that no combustible escapes oxidation. Such feature here designated as a furnace, if of a length of about 14 feet, will produce an ideal mixture of the gases if the coal is supplied to the grate at a uniform rate, but, if not, its ability to effect mixture of the gases is not sufficient for complete combustion. With the thick fire of a chain grate stoker feeding bituminous coals, a considerable amount of combustible gas escapes from the front of the fire, such a quantity, in fact, that an ignition arch alone, although 5 feet in length, does not cause sufficient mixture of the gas and air to produce complete combustion. Therefore, if the maximum requirement of the furnace is to thoroughly mix together the gases from a chain grate fire, then the efficiency of such furnace feature as fitted under boilers *A* and *B* is ideal, because proper regulation of the quantities of air and fuel supplied by the grate, resulted in the production of 18 per cent. CO_2 with complete combustion and no smoke.

17. The performance of the chain grate stokers under boilers *A* and *B* are unsatisfactory in two respects: First, that with desirable strength of draft the available thickness of fuel bed does not supply sufficient combustible to satisfy the air supply. Second, fuel is wasted by passing over the end of the grate with the ash. This latter fault may be remedied by running the grate at a speed which will allow the fuel to burn away before it reaches the end, but this results in increased excess of air, therefore the chain grate stoker, in combination with the above described furnace feature, is inefficient to the extent by which it fails to supply a sufficient thickness of fire or quantity of combustible, and the loss of fuel with the ash it not a fault of the stoker, because if the

supply of combustible was sufficient to satisfy the air the grate could be run at a speed which would insure that the fuel all be burned.

18. It is the author's wish in presenting this paper to suggest the advantages of more detailed study of steam generating apparatus. Such results as shown by boiler *B* over *A* would justify considerable investigation in connection with either selection or design, and the burning of inferior bituminous coal without smoke is a feature of increasing value.

DISCUSSION

Prof. Wm. Kent.—If it is desirable to have two boilers tested under identical circumstances so that each shall be supplied with the "same relative quantity of heat at the same temperature," and this can probably be done most easily with natural gas, both the gas and the air being supplied from meters. It would be almost impossible to secure identity of conditions with coal fuel. Suppose it were possible to make such tests, what would be the use of them? They would probably show that the plain cylinder boiler has a greater efficiency for a given rate of evaporation per square foot of heating surface per hour than any other boiler, provided that the radiation loss from this boiler could be reduced to as low a percentage of the total heat generated as it is in other boilers. This result would follow from the fact that the plain cylinder boiler is about the only one that has ever been made in which there is no possibility of some short circuiting of gases; and of what use would this result be if it were found? It would not lead to anyone's prefering to use a plain cylinder boiler to any other kind of boiler, for it has too many defects which overbalance its single point of superiority.

The statement made in paragraph 8 of the paper that the efficiency of any particular boiler is a fixed quantity which once ascertained according to some standard applies to any number of reproductions and thus need only be determined once may also be true, but of what use is it if it is practically impossible to fix a standard set of conditions which can be duplicated on two successive days. It might possibly be duplicated if the conditions of firing were natural gas supplied from a meter at a definite rate per hour and air supplied also at a definite rate and of a definite condition as regards dryness, and so mixed with

the gas that complete combustion takes place before the products of combustion touch the heating surface. If such a standard set of conditions could be obtained this standard would have to vary for each kind of boiler and for each variation in shape or dimensions of each boiler in order to obtain its highest efficiency. For instance, if radiation of the horizontal cylinder could be suppressed, it might be possible to obtain an efficiency of 80 per cent. at a rate of driving of 4 pounds of water per square foot of heating surfce per hour after having experimented long enough to find what mixture of gas and air would give the highest furnace temperature. Then experimenting with other boilers, it would be found that the same efficiency of 80 per cent. could be reached with the same analysis of the gases of combustion at rates of driving varying with each kind of boiler and with different sizes and shapes of the same kinds from two pounds to something less than four pounds. And what would be the use of such a result if we had it, if the efficiency of these same boilers tested with coal would vary with no sort of regularity and according to no ascertainable law with every variation in kind of coal and character of firing? For example, I once tested a water-tube boiler in which the gas passage were arranged exactly as in boiler *A* described in the paper. At a moderte rate of driving I got a very low efficiency and a high temperature of the furnace gases. I attributed the poor results to short circuiting, and said that next day I would drive the boiler higher, would obtain higher capacity, higher efficiency and lower chimney temperature, which I did. An entirely different set of results might have been obtained if a different kind of fuel had been used. Of what importance was it for me to obtain a definite figure for the efficiency of this boiler under a "standard" set of conditions such as burning, say, anthracite egg coal, with a definite force of draft and a definite rate of combustion per square foot of grate surface, if the boiler was actually to be used with bituminous coal which varied in quality every day in the week?

The author states that a furnace having a refractory roof and length of fourteen feet will produce an ideal mixture of the gases if the coal is supplied to the grate at a uniform rate, but if not its ability to effect the mixture of the gases is not sufficient for complete combustion. This statement is no doubt true for a certain grade of coal fired at a certain rate, but if we tried lignite at the same rate or the same coal supplied to the grate at

double the rate but yet a uniform rate, it is not probable that fourteen feet length of the furnace will be nearly sufficient to cause complete combustion unless other means are taken to cause mixture of the combustible gases and air in the furnace. Such a means of mixture is provided in my Wing Wall furnace, and such a means also is provided in under feed stokers, in downward draft furnaces and in chain grate stokers when supplied with a sufficient length of arch and run in such a manner as to supply enough air through the rear portion of the grate to burn the gases produced in the front portion.

Mr. Bement's paper contains some interesting information and deductions, but it does not lead to any more exact methods for determining the efficiency of steam generating apparatus than we now have. Our methods of determining the efficiency are already about as exact as they need be. What we need is some means of controlling the efficiency, and that will not be supplied until the fireman is furnished with a registering pyrometer which will be as easy to read as the steam gauge is already, and apparatus which will indicate continuously the percentage of oxygen in the furnace gases.

Prof. E. A. Hitchcock.—I trust that Mr. Bement's suggestions will bring out others along this line with experiences, attempts, suggestions and failures, so that some who are constantly making investigations along steam generation lines may profit thereby, and not start out on paths that will have to be retraced.

If boiler efficiency is to be determined by the condition and temperature at the furnace of the products of combustion, the fields of trials at present will be quite limited, for to draw correct comparisons between different types of boilers they should have the same type of furnace, one that will give complete combustion of gases with high temperature before the boiler surfaces are reached. The temperature of the gases leaving the furnace would, of course, be very important, since two boilers identical in type, size and condition and, therefore, of the same efficiency, would show different efficiencies with different furnace temperatures, other conditions remaining the same. This requirement would eliminate all boilers where the grate is in close proximity to the heating surface, and also all of the internal fired type. It would also be very difficult with the latter to obtain the temperature and a sample of the gas for analysis just after leaving the furnace, requirements necessary in all cases. My experience in

obtaining furnace temperatures and a continuous sample of gas just before entering the tubes on a Babcock & Wilcox boiler with Dutch oven furnace, has been that it is difficult and expensive. I should say that every boiler should have a furnace that will give, when properly handled, practically complete combustion with high temperature. Then a complete boiler heat balance trial will at once show if the boiler is being handled properly for the conditions existing. That is what the owner of the boiler plant wants to know.

Now, if Mr. Bement had obtained complete heat balances for tests on boilers *A* and *B*, Table II, we could see at once why there was a saving in fuel of 20.68 per cent. Now, assuming, as was probably taken for granted, that all the conditions of each test was constant, the only difference being in the final temperature of the escaping gases, then the gain was due entirely to the reduction in temperature of the products of combustion and air in excess. Now, boiler *A* absorbed $13{,}633 \times 0.5293 = 7{,}220$ British thermal units, and *B* $13{,}633 \times 0.6673 = 9{,}110$ British thermal units per pounds of coal, or a difference of 1,890 British thermal units, with a drop in temperature of the products of combustion of *B* over *A* of 152 degrees, which gives 12.4 British thermal units per degree of temperature. As the products of combustion on boiler *A* had a temperature of 514 degrees above the air supply, the loss, therefore, in that direction, on a basis of 12.4 British thermal units per degrees, would be $12.4 \times 514 = 6{,}373$ British thermal units per pounds of coal. Now, $(7.47 \times 965.7) + 6{,}373 = 13{,}586$ British thermal units, leaving a difference of only 47 British thermal units, or 0.3 of one per cent. to be charged to two sources of loss at the very least, that is, radiation and unburned coal in refuse. There must have been some loss due to unburned coal, especially since the boilers were equipped with the chain grate stoker, and there is certainly loss in radiation. To have a loss in the products of combustion of 6,373 British thermal units per pounds of coal, with a temperature of products of combustion above atmosphere of 514 degrees, would mean an air excess of about 450 per cent. These points seem to indicate that in the two trials given the conditions were not the same; in fact, the drafts above the fires are quite different which would unquestionably make quite a difference, the kind of coal and thickness of fires remaining the same. I consider a boiler trial incomplete when there is no heat balance, and different trials on

the same, or different boilers, are incomparable without it. I cannot agree with a prominent engineer who recently stated in print that a boiler heat balance looks nice on paper, but practically was of absolutely no value. If as much time, thought, experimentation, etc., had been spent during the past on the steam generation side of the power plant as on the power side we would to-day have more efficient systems.

Mr. Arthur Herschmann.—The word "circulation" stands for efficiency in steam raising. The water should flow unchecked from low to high temperature until it leaves the boiler as steam, while the furnace heat should be communicated to the boiler contents until its temperature is lowered to a certain minimum, when the gases should be allowed to escape.

I wish to emphasize the well-known fact that with the average boiler plant the flue gases escape at an excessively high temperature. The reason is that the stack would not produce draught strong enough to supply the requisite amount of air for combustion were it otherwise.

A very high stack—on the other hand—which produces strong draught with little "head of heat," as we might say, would be wasteful of heat whenever a fixed relation between the position of its damper and the condition of the fire is disturbed.

For the above reasons it would seem that "blowing air through the fire" is deserving of more careful consideration, and that it should be made the subject of accurate comparative tests to determine its value with high stacks, such as we have in modern office buildings. Efficiency is raised by blowing air so as to produce a balance of pressure above the fire and helping the products of combustion being drawn through the damper, which latter is set to fit conditions.

Mr. George H. Barrus.—The title of this paper is a misnomer. In the closing paragraph, the author says that its object is merely to suggest advantages of more detailed study of steam generating apparatus, but it might be inferred from the title that its object is to criticize the existing methods for determining the efficiency of boilers. I have carefully read the paper and I do not find in it any such criticism, neither do there appear any suggestions for improving the methods already well-known and thoroughly embodied in the report of the work done by the Boiler Test Committee of the society. It may be inferred from reading the title "More Exact Methods of Determining the Efficiency of Steam

Generating Apparatus" that the methods established by the society's committee are not sufficiently exact, and that the author has better ones. There is, however, no contention of this kind. Considering that the paper merely advocates a more complete analysis than is sometimes undertaken, the wording of the title should be changed accordingly.

I object to the use of the words "pure coal" in Table number 2, instead of the commonly accepted term used in this country, "combustible." The term "combustible" has come to be well known by long usage, and it has been incorporated in all the tables of the various testing committees of the society which have passed on these subjects. It is true that the word "combustible" does not quite fill the bill, but it is equally true that the term "pure coal" has a like failing. Individually, I consider that "combustible" is a far better name than "pure coal," and there is no advantage whatever in substituting the less expressive term.

It would increase the value of the paper if sketches or drawings were added, showing the leading features of the two boilers *A* and *B*, to which extended reference is made. From the reading of the description, I am unable to fully understand the arrangement, and no doubt others may find the same difficulty.

Mr. Albert A. Cary.—I will be obliged to take exception to the first paragraph of Mr. Bement's paper.

If a boiler test be carefully conducted in accordance with the complete form advised by the American Society of Mechanical Engineers' Boiler Test Committee in the Code of 1899, sufficient information will be obtained to determine the separate and individual efficiencies of both the boiler and the furnace which, as correctly stated by Mr. Bement, are necessary to obtain exact information concerning steam generating apparatus.

By referring to my discussion of the "Revision of the Code for Conducting Steam Boiler Trials" at the December 1899 meeting of this society, it will be seen that I suggested the necessity for a statement of boiler efficiency apart from the furnace efficiency, and I can state now that I am a much firmer believer in the necessity of the separation of these two efficiencies than I was five years ago.

Most steam users seem to regard the furnace and boiler as one integral piece of apparatus which is wholly wrong. They are as separate one from the other as the boiler and the engine.

The furnace is purely and simply a chemical apparatus, while the boiler is a physical apparatus, and in testing they should be treated as such.

The furnace, by chemical action, is a *producer* of heat, while the boiler (diametrically the opposite) is an *absorber* of a portion of such heat as the furnace delivers to it.

Any material advance in the improvement of steam generating apparatus cannot proceed until it is thoroughly appreciated that both boiler and furnace have their own individual efficiency, and the product of the two separate efficiencies will give the *combined* boiler and furnace efficiency, which latter efficiency, as has been indicated by Mr. Bement, has been too commonly accepted without proper analysis to determine whether we are operating with a boiler of high efficiency connected to a furnace of low efficiency, or whether our boiler is very wasteful while the furnace is all that it should be. When such an analysis is made, it is often astonishing to see how quickly defects are discovered which may, in many cases, be remedied and followed by a decrease in fuel consumption and by a suppression of the smoke nuisance.

Furnace efficiency may be defined as the ratio of the total heat delivered from the furnace to the total heat value of the coal consumed.

Boiler efficiency may be defined as the ratio of the total heat absorbed by the boiler to the total heat delivered by the furnace to the boiler.

With this preface I will now proceed to show how boiler and furnace efficiencies may be arrived at by use of the information obtained in a boiler test conducted according to the complete form of the American Society of Mechanical Engineers' Code.

The following table gives the heat balance of a combined furnace and boiler test I conducted, which will show how the heat is distributed between the furnace and boiler:

HEAT BALANCE OR DISTRIBUTION OF THE TOTAL HEAT CONTAINED IN THE COAL.

TOTAL HEAT VALUE OF 1 LB. OF COAL, AS DETERMINED BY MAHLER BOND CALORIMETER, 12,559 BRITISH THERMAL UNITS.

	B.t.u.	Per cent.
(*a*) Heat absorbed by boiler as found in dry steam evaporated per pound of coal	8,019	63.850
(*b*) Loss due to non absorption of heat contained in the steam evaporated from the Moisture in the coal / " " " air	229	1.820
(*c*) Loss due to the non-absorption of heat contained in the steam formed by the burning of hydrogen	36	0.283
(*d*) Loss due to heat carried away in gases delivered to chimney: Heat in the CO_2-CO-O-N, etc. / " " " moisture from the coal / " " " " " burned hydrogen / " " " " " the air	2,121	16.886
(*e*) Loss due to incomplete combustion of carbon	876	6.972
(*f*) Loss due to unconsumed hydrogen and hydrocarbons	580	4.628
(*g*) Loss due to unburned carbon leaving the furnace with the ash	270	2.150
(*h*) Loss due to heat withdrawn from furnace with ash	26	0.211
(*i*) Loss due to radiation	402	3.200
Total	12,559	100.000

We can now, with the heat balance before us, separate our charges against boiler and furnace, and so arrive at our boiler and furnace efficiencies.

This balance shows charged against the furnace:

	B.t.u.	Per Cent.
Item (*e*)	876	6.972
" (*f*)	580	4.628
" (*g*)	270	2.150
" (*h*)	26	0.211
Total	1,752	13.961

This gives us a furnace efficiency of $(100 - 13.961) = 86.039\%$.

The following losses are chargeable against the boiler:

	B.t.u.	Per Cent.
Item (*b*)	229	1.820
" (*c*)	36	0.283
" (*d*)	2,121	16.886
" (*i*)	402	3.200
Total	2,788	22.189

From these results it will be seen that the boiler efficiency referred to the total heat value of the coal is

$$(100 - 22.189) = 77.811\%.$$

But we know that the total heat value of the coal is not delivered to the boiler by the furnace, there being only 86.039% of this amount of heat available for use by the boiler. By simple proportion we have:

$$22.189 : 86.039 :: X : 100.$$

the value of X being 25.79%, which represents the true loss charged against the boiler. The true efficiency of the boiler is therefore $(100 - 25.79) = 74.21$ per cent.

To check this boiler efficiency result we can use the well-known therodynamic formula:

$$100\,\frac{H - h}{H} = E,$$

In which

H = The British thermal units delivered by the furnace to the boiler (*i.e.*, 12,559 − 1,752 = 10,807 British thermal units).

h = The British thermal units rejected from the boiler to the chimney and from the boiler by radiation (*i.e.*, 2,121 + 229 + 36 + 402 = 2,788 British thermal units).

E = The efficiency of the boiler referred to the total heat delivered by the furnace to the boiler.

By substitution we have

$$100 \times \frac{10{,}807 - 2{,}788}{10{,}807} = 74.21 \text{ per cent.}$$

which is the same result as obtained before.

By taking 74.21 per cent. of 86.039 we find that combined efficiency of the boiler and furnace is $(74.21 \times 86.039) = 63.85$ per cent., the same as shown in item (a) of our heat balance.

With an initial temperature in the furnace, varying between 2,500 degrees and 3,000 degrees F., as existed during the test reported above, a boiler efficiency of 74 per cent. is too low, whereas the furnace efficiency of 85 per cent. is good, although a somewhat higher efficiency is practically possible. In another boiler of the same type, after making temperature tests and gas analyses, I was able to make changes which raised the boiler from a similar efficiency to that just given to an efficiency of nearly 85 per cent. If the boiler we have been considering had an 85 per cent. efficiency, the combined efficiency of boiler and furnace would have been (85 per cent. × 86 per cent.) = 73.1 per cent. which would mean a saving of 9.25 per cent. in the fuel bill.

Conducting an elaborate and careful combined boiler and fur-

nace test, according to the American Society of Mechanical Engineers' Code, such as is necessary to make a heat balance as given above, entails altogether too much work should we merely wish to ascertain the individual furnace efficiency.

I have by another and more simple method of procedure succeeded in obtaining this information, but as a description of this method would carry us away from a proper discussion of this paper, I will not attempt to describe it here, but will make it the subject of a paper to present at the coming meeting of the society.

In my work of designing and improving furnaces, I do not always find a boiler attached to the furnace, and therefore such a procedure as outlined above becomes impossible, but by the special method—just referred to—the furnace efficiency can be ascertained with equal accuracy whether the furnace is used for metallurgical purposes, for burning cement or for any other purpose.

Referring now to paragraphs 4 and 5 of Mr. Bement's paper we see, with the discussion I have just presented, that it is not necessary to establish a set of uniform conditions requiring the delivery of the same relative quantity of heat at the same temperature in order to arrive at this information regarding the individual efficiencies of furnace and boiler, although, of course, it is desirable where a comparison of two different types of boilers and their standard settings is wanted, to use standard boilers of as nearly the same rated horse power as possible in connection with furnaces built as nearly alike as conditions will permit, and of course, it is well in such comparative tests to use the same coal and have the fires handled as nearly alike as possible.

As Mr. Bement has stated, the individual efficiency of any particular boiler is a fixed quantity, and when once determined will apply to any number of reproductions, providing the size, setting, travel of gases and length of time that they are in contact with all parts of the heating surface are identical, but the breaking down or displacing of baffles or any other cause which would produce a short circuiting of gases, will decrease the efficiency of the boiler.

For a matter of reference, all individual efficiencies of boilers should be stated with some standard assumed temperature of gases delivered from the furnace, as well as with some standard assumed temperature of gases discharged by the boiler to the flue outlet, and factors may be determined which when multiplied by

the individual boiler efficiency, as just described, will give the change of efficiency in the boiler as the temperature of the furnace and flue gases change.

Such factors could be arrived at with comparative ease if the specific heat of the hot furnace gases was a constant quantity at all temperatures, but the change in the specific heat of gases at different temperatures is, unfortunately, a matter we know too little about at present.

I can endorse the assertion made in paragraph 10 of Mr. Bement's paper, that there is probably a greater difference in individual boiler efficiencies than is commonly appreciated.

I do not mean that the value of a square foot of heating surface in any one boiler is greater than in some other boiler per se., but my experience has shown that the travel of hot gases from the furnace to the flue outlet is not as well regulated in some boilers as in others.

Such gases should be as uniformly distributed as possible over the entire heating surface as is consistent with proper draft conditions, and within practical limits, the greater the length of time the moving gases are held in contact with the heating surface of the boiler, the greater will be the boiler's efficiency.

These facts have been made most apparent in a number of tests I have conducted.

In one type of boiler in which the travel of gases from the furnace to the flue outlet is of considerable length, but in which, with the standard method of setting, the furnace is not an ideal one for the use of bituminous coal, with such fuel the percentage of boiler efficiency is always higher than the percentage of furnace efficiency.

With another type of boiler, in which the length of travel of the furnace gases is less than with the other boiler just mentioned, and in which, with standard setting, the furnace is well adapted for the use of bituminous coal—with such fuel I have always obtained a higher percentage of furnace efficiency than percentage of boiler efficiency.

I have found in many cases that the loss in one direction in one of these type of boilers has offset the loss in the other direction in the other type of boiler, so that the combined efficiencies of the two boilers have shown about the same results.

This shows us the value of separating the furnace and boiler efficiencies, as with such information I have been able to change

the construction of the poorly efficient furnace attached to the highly efficient boiler and thereby obtain a materially higher combined efficiency and fuel saving, to say nothing of the smoke suppression, while in a certain case where the well constructed furnace was used with the less efficient boiler, I was able, by making a better disposition of the furnace gases, to raise the boiler efficiency and obtain an excellent combined efficiency and fuel saving.

If this matter of individual boiler and furnace efficiencies had been better understood during the past few years, we would not see so many bad examples of steam plant practice as exist to-day, in which the proper balance between furnace and boiler has seemed to have been lost sight of.

In large plants this bad practice has become almost epidemic, and they are paying for it dearly, while the coal companies are reaping a splendid harvest.

I have in my practice found confirmation of the results shown in Mr. Bement's tables No. 1 and No. 2, which are well worth careful consideration, as they show avoidable losses which exist in many steam plants.

As regards the use of the expression "pure coal" in preference to "combustible," I believe this to convey a better meaning of what is intended to be conveyed by its use.

Mr. Snow uses this expression in his work on Mechanical Draft, and Mr. Pool uses it in his work on Fuels.

Mr. Bement's statement that a refractory arch over the furnace and combustion chamber of a length of fourteen feet will produce ideal conditions, is probably made in reference to Northern Illinois and Indiana coals, but there are other matters of equal importance affecting furnace efficiency, some of which were referred to in my topical discussion at the Saratoga meeting of this society.

Prof. D. S. Jacobus.—This paper appears to propose that the efficiency of a boiler be based on the amount of heat it absorbs, as computed from the temperature of the flue gases. It would be impossible under ordinary conditions to secure exact results in this way. In the first place, the exact temperature of the flue gases is hard to determine. In some boilers there are differences of 50 to 100 degrees or more in the reading of a thermometer when the bulb is held in various positions in the flue. Even if the temperature of the flue is determined at various points it is hard

to say how to obtain a correct average because the velocity of flow of the gases varies at different parts of the flue. Furthermore, if the efficiency is to be computed in this way it would be necessary to determine and allow for the amount of air leakage in the boiler setting. It therefore appears that the method proposed is subject to a much greater error than the one ordinarily adopted, and for this reason its use should not be recommended.

Prof. Wm. Kent.—I have heard a great deal about this separating the efficiencies of the furnace and boiler, and I am not quite sure about it, and I hope Mr. Cary when he revises his discussion he will make it clear. Imagine a water-tube boiler under which we are burning western bituminous coal high in moisture. We may have a temperature of 1800 degrees in the furnace and the same temperature in the chamber above the tubes, the flame continuing all the way, and a temperature at the flue of 1000 degrees. The next day we change the conditions, and we have a different coal, say dry anthracite, and by very careful firing and careful proportion of draught and air supply we may get a temperature of 3000 degrees in the furnace, 1500 degrees above the tubes, and 500 degrees at the flue. The next day we test it again with anthracite coal and we have a great excess of air so that the temperature is only 2000 degrees in the furnace, with an efficiency of 100 per cent, 1500 degrees above the tubes, and say 700 at the flue. Now the efficiency of the boiler differs in the different cases, and the efficiency of the furnace is 100 per cent. in two cases of perfect combustion. What is it in the first case, where the combustion is incomplete, and should the low efficiency of the boiler be charged to the furnace or to the unfavorable conditions?

Mr. Albert A. Cary.—Concerning the variation of boiler efficiencies with a change of furnace and flue outlet temperatures, referred to by Prof. Kent.

I have already considered this in my discussion, though with somewhat less elaboration, and suggested a means of expressing boiler efficiencies so as to eliminate the difficulties he had described.

Concerning the matter of accounting for air leakages around the settings of boilers referred to by Professor Jacobus, I have accounted for such infiltrating air quite satisfactorily by collecting simultaneous samples of furnace gases at the furnace and at the flue outlet, and the difference between the excess of air ac-

counted for at these two places by gas analysis give the amount of air passing through the setting.

Mr. George I. Bouton.—In paragraph 14 Mr. Bement tells us some of the conditions under which this test was made, and they evidently made an attempt to get uniform conditions under both boilers *A* and *B*, but they also evidently overlooked the fact that by inserting the baffles referred to they did practically the same thing as they would do if they shut down the damper in boiler *B* so much they only got 0.22 draught, while in boiler *A* they have 0.38. Now, by referring to Mr. Bement's paper, read before the Western Society of Engineers some time ago, you will find a table in which he gives the variation in efficiency of a boiler equipped with chain grate stokers with a variation of draft, and you will find that table accounts for over half of the difference in efficiency (shown in table No. 2) as due to the difference in draft. They were apparently getting all the air they could through the grate, due to the draft that was available, so that in the tests referred to in Table No. 2 they must have been getting a very large excess of air, because they were only burning about two-thirds of the coal burned in tests shown in Table No. 1, where they show about 10 per cent. CO_2, so the temperature must have been very low and the efficiency shown was not what it would be if the dampers had been properly regulated.

*Mr. A. Bement.**—Prof. Kent reverses the matter from the standpoint at which I present it, and refers to the production of a standard efficiency, thus making capacity the measure of superiority, and asks what would be the advantage of knowing that under such conditions one boiler evaporated two pounds and another something under four pounds of water per hour per square foot. The answer is that a purchaser could select that with the largest capacity, and in so doing need only purchase a little more than half as many boilers, real estate to place them on and buildings with which to cover them, than he must if selecting that boiler of the two pound square foot capacity. As to anyone preferring one kind of boiler over another on account of its possible advantage, I would say that they no doubt would, because certain boilers are now very often bought on account of the belief that they are more economical than others, and if a superior efficiency was known, it would cause people to prefer such boilers.

* Author's Closure, under the Rules

Prof. Hitchcock has apparently overlooked the statement that the weights given in Table No. 2 are the pounds of coal burned. The amount fed by the stokers was greater, and there was loss of fuel in the refuse. These losses were determined by weighing the refuse, also sampling down the entire quantities and analyzing these samples. So that the quantities of fuel not burned were subtracted from the quantities which were fed, and the calculations were based on the pounds burned, and the word "burned" was used with no reservation, because combustion was complete in each case. This was not a test of the entire apparatus, but of the boilers only, and the amount of coal burned, and equivalent evaporation per pound of fuel was very easily and accurately determined, and a heat balance with its difficulties and complications was not required to determine the result.

The 0.3 per cent. of the heat as calculated by Prof. Hitchcock for boiler *A* should have only the radiation charged against it, and if that quantity is not sufficient, it illustrates one of the difficulties in making a heat balance. As stated in the paper, the condition of combustion was not good, inasmuch as the excess of air was very large. CO_2 for furnace of boiler *A* being 7.0 per cent., and for *B*, 6.7, the lower CO_2 of the latter was the product of two influences. Furnace *B* burned much less coal, therefore the tendency was for the fixed leakage of air to be proportionately more than with the other furnace, but as furnace *B* had a smaller opening at the bridge-wall, this condition very nearly neutralized the other, therefore, as far as effect on the boilers was concerned, it was slightly in favor of *A*. The coal used in these tests was not selected, but was what came down from the bunkers at the time, and owing to its size, and to the fact that the stokers are faulty in some respects, as mentioned in my paper, a better condition of combustion could not be secured at the time. But I am quite willing to make a heat balance, and extending Prof. Hitchcock's analysis, we have Table No. 3, in which the radiation loss, while appearing a little lower than would probably be expected, is entirely consistent. Slightly lower for boiler *B*, as it should be, not because this boiler radiated less heat, but because the horse power produced through it was greater than with the other.

The fact of less draft in furnace *B* is not an indication of difference of conditions, as far as the object of the test is concerned, because it was caused by the boiler itself owing to the greater

TABLE 3.

DISPOSITION OF THE HEAT; BOILERS *A* AND *B*.

	BRITISH THERMAL UNITS.		PER CENT.	
	A	*B*	*A*	*B*
In making steam	7,220	9,110	52.93	66.73
Lost in escaping gases	6,373	4,489	46 73	33.02
" " radiation by difference. ...	40	34	0.34	0.25
Total	13,633	13,633	100.00	100.00

resistance of the complicated gas passages in it and not from any other cause. The effect of reduction of draft with this furnace was to burn less coal than with the other, but the fact should not be overlooked; notwithstanding this, its boiler made the greatest horse power. Table No. 4 will illustrate these features:

TABLE 4.

UNIFORM CONDITIONS FOR BOILERS *A* AND *B*.	DIFFERENT RESULTS PRODUCED BY BOILERS.		
		A	*B*
Draft at damper.	Draft at fire	Greater.	Less.
Steam pressure and temperature.	Quantity of coal burned	Greater.	Less.
Extent of heating surface.	Steam made per pound of coal	Less.	Greater.
Condition of combustion (approximate)	Horse-power produced..	Less.	Greater.
Quantity of heat lost by radiation.	Heat lost to the chimney	Greater.	Less.
	Percentage of heat lost by radiation..	Greater.	Less.

I am in a measure disposed to agree with the remarks made by Mr. Barrus regarding the title of my paper; it would have been more to the point if I had intimated therein that we had no method of testing boilers and are in need of one.

In answer to Prof. Jacobus, it was not my intention to suggest that the efficiency of a boiler be based on the amount of heat it absorbs as computed from the temperature of the flue gases. My remarks in that connection were offered as introductory illustrations. My suggestions were for certain standards for the influencing conditions, namely, steam pressure, condition of combustion, and rate of evaporation per unit of boiler surface, the result to be computed by referring the heat supplied to the heat absorbed, as

$$E = \frac{\text{Heat absorbed per pound of fuel burned}}{\text{Heating value of one pound of fuel burned}}$$

just what we do now, but which gives various incorrect results, due to variation in influences, other than of the boiler itself.

Mr. Bouton is mistaken in considering that the diagram to which he refers offers any explanation of the performance of these boilers other than that which I have given. If he will study the matter he will see that this is so.

Inasmuch as the valuable discussion of this paper has brought out important points which cannot have attention in these closing remarks, I shall prepare another paper for presentation at the next meeting of the society.

PAPERS

OF THE

SCRANTON MEETING

(LIst)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

JUNE 6th TO JUNE 9th, 1905.

No. 1075.

PROCEEDINGS

OF THE

SCRANTON MEETING

(LIst)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

June 6th to 9th, 1905.

LOCAL EXECUTIVE COMMITTEE.

WM. F. MATTES, *Chairman.*
RUFUS J. FOSTER, *Chairman Local Finance Committee.*
A. B. CLEMENS, *Secretary General Local Committee.*

The fifty-first meeting of the American Society of Mechanical Engineers was appointed for the city of Scranton, Pa., by invitation of the members of the Society in that city, the Board of Trade of Scranton, and the Scranton Engineers' Club.

The opening session was called to order on the evening of Tuesday, June 6th, 1905, at half-past eight, by Mr. Wm. F. Mattes, Chairman of the Local Committee, who called attention to the fact that the Society had met in Scranton in October, 1888, for its eighteenth meeting, and that now, after seventeen years, he had again the honor of being chairman of the local committee of arrangements to welcome the visitors.

In the unavoidable absence of Mayor Connell of the city, Colonel F. L. Hitchcock of the Scranton Board of Trade presented to the Society the welcome on behalf of the citizens. Mr. E. M. Zehnder, President of the Scranton Engineers' Club, presented the greetings and hospitality of a young and active engineering

organization to the visitors, and in fitting response President John R. Freeman replied on behalf of the Society. President Freeman then took the chair and opened the session for the transaction of business.

Invitations were read from the Scranton Club and from the general secretary of the Young Men's Christian Association, in whose auditorium the meetings were held. Professional papers were then taken up. Attention was called by the Secretary, in presenting the first paper, to the fact that by coöperation of authors with the effort of the Secretary's office it had been possible to comply fully with the provisions of the rules which called for the transmittal of the entire batch of papers to be read at any one meeting, from his office, thirty days in advance of the date set for the meeting itself.

This possibility and practice had resulted not only in securing discussion in writing from members resident at considerable distance, and even from across the Atlantic, but had also made it possible to have many of these written discussions set up in type and in form for distribution at the meeting, with the manifest advantages to all participants in debate and to the authors themselves.

The first two papers, by Mr. F. C. Wagner, "The Transfer of Heat at High Temperatures," and Mr. F. E. Matthews, "Standard Unit of Refrigeration," received no discussion.

The paper by Mr. Howard T. Barnes, which had been presented at one of the local reunions in the City of New York during the winter, entitled "Formation of Anchor Ice and Precise Temperature Measurements," was discussed by Messrs. Freeman and Mattes.

The concluding paper of the session was that of Mr. Wm. O. Webber, "Some Types of Centrifugal Pumps," discussed by Messrs. Freeman, Henshaw, Lewis, Mattes, Morse, Ray, Rearick, Rice, Suplee, Trump, Wilson, and Zohe.

At the close of this discussion, after announcements by the Secretary and invitations not included in the regular printed list, a recess was taken until the following morning.

SECOND SESSION. WEDNESDAY MORNING, JUNE 7TH, 10 O'CLOCK.

This session was the business session of the meeting. The Chairman called for the report of the Tellers of Election, which was read for record by the Secretary as follows:

REPORT OF THE TELLERS OF ELECTION.

The undersigned were appointed a committee of the council to act as tellers under by-laws 6, 7 and 8, to scrutinize and count the ballots cast for and against the candidates proposed for membership in their several grades in the American Society of Mechanical Engineers, and seeking election before the fifty-first meeting, Scranton, Pa., 1905.

They have met upon the designated days in the office of the Society, and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society, to the election of the following persons, whose names appear on the appended list in their several grades.

There were 589 votes cast on the ballot ending March 15, 1905, of which 51 were thrown out on account of informalities. There were 657 votes cast on the ballot ending May 29, 1905, of which 53 were thrown out because of informalities. The tellers have considered a ballot as informal which was not endorsed, or where the endorsement was made by a facsimile or other stamp.

CHARLES E. LUCKE,
ALBERT SPIES,
HENRI G. CHATAIN,
} *Tellers of Election.*

ON THE

FIRST BALLOT FOR MEMBERSHIP,

APPROVED BY COUNCIL ON JANUARY 31, 1905

Which Ballot closed March 15, 1905.

MEMBERS.

Abbott, C. C.	Hall, F. B.	Moore, F. C.
Baker, D. G.	Hubert, H.	Moyer, H. C.
Carlsson, C. A. V.	Kahn, J.	Prince, W. F.
Clemens, C. W.	Leach, W. H., Jr.	Richards, F. W.
Crocker, A. S.	Montague, C. D.	Robinson, J. R.
Starr, J. E.		Thullen, L. H.

PROMOTION TO FULL MEMBERSHIP.

Braine, B. G.	Cole, L. W.	Jarecki, A. H.
Bush, H. M.	Earle, E. P.	Peck, C. B.
Cheney, H. W.	Funk, W. F.	Quimby, W. E.
	Sayers, W. W.	

ASSOCIATES.

Clark, E. L.	Price, M.	Stehlin, J.
Ostrander, A. E.	Schneller, G. O.	Willhofft, F. O.

PROMOTION TO ASSOCIATE MEMBERSHIP.

George, J. Z.

JUNIORS.

Alsberg, J.	Earle, S. B.	Hutchins, H. C.
Barnes, C. B.	Hamilton, T. S.	Kent, R. T.
Baylis, A. R.	Harrison, E. S.	Northrup, F. B.
Been, P. H.	Helvey, C. H.	Rauch, J. D.
Brooks, P. R.	Hirshfeld, C. F.	Shiebler, M.
	Smith, E. B.	

All elected.

HENRI G. CHATAIN,
CHARLES E. LUCKE.

ON THE

SECOND BALLOT FOR MEMBERSHIP,

APPROVED BY COUNCIL ON APRIL 25, 1905,

Which Ballot closed May 29, 1905.

MEMBERS.

Austin, W. F.	Gowie, W.	Pingree, E. D.
Bailey, F. W. C.	Hart, F.	Rapley, F. H.
Baldwin, C. K.	Horton, S. E.	Roper, N. B.
Bennett, G. L.	Jackson, C. J.	Ruggles, W. B.
Browning, C., Jr.	Jacobs, H. W.	Schlesinger, G.
Burleigh, C. B.	Johnson, P. F.	Sherrerd, J. M.
Burleigh, W. F.	Kunhardt, L. H.	Shipman, R. L.
Carroll, A. W.	Long, J. C.	Town, F. E.
Coleman, E. P.	Martin, F. S.	Trinks, C. L. W.
Crabtree, F. H.	Naylor, C. W.	Tucker, F. S.
Cummings, W. W.	Nettleton, W. A.	Wait, H. H.
Dean, E. W.	Ord, H. C.	Wells, G. A., Jr.
Fry, L. H.	Parker, J. C.	Whiton, L. E.
	Williston, B. T.	Zimmerman, O. B.

PROMOTION TO FULL MEMBERSHIP.

Bailey, T. S.	Girvin, C. J.	Herschmann, A. J.
Barnard, W. N.	Glanville, J. G.	Hinchman, T. H., Jr.
Bird, J. D.	Hall, R. E.	Lindquist, E. W.
Echeverria, R. J.	Herron, J. H.	Murray, H. H.
Norris, J. H.		Wilkinson, T. L.

ASSOCIATES.

Berry, E. H.	Carrier, W. H.	Jurgensen, J. C.
Bridge, J. W.	Einstein, A. C.	Prince, J. W.
Bronaugh, W. L.	Flinn, C. F.	Sanders, L.
	Schumaker, J. S.	

PROMOTION TO ASSOCIATE.

Van Valkenburgh, R. D.

JUNIORS.

Allen, W. C.	Green, C. H.	Marsh, T. A.
Austin, A. O.	Griffiths, L. L.	Murphy, B. S.
Bacon, C. J.	Hamilton, E. W.	Murphy, E. T.
Barstow, F. L.	Haney, J. B.	Murrie, J. L.
Buckler, A. H.	Horne, C. F.	Myers, C. C.
Church, H. B.	Jones, D. L.	Northrup, L. M.
Comly, G. N.	Keith, T.	Owen, I. J.
Cooke, S. G. H.	Lang, C.	Palmer, V. M.
Cox, F. G.	Lathrop, W. F.	Proctor, R., Jr.
Cushman, A. W.	Lucas, H. Van Noye	Smith, E. B.
Fleming, W. M.	McGregor, A. G.	Weeks, P.
Wheeler, S., Jr.		Whipple, W.

All elected.

CHARLES E. LUCKE.

On behalf of the Council the Secretary presented an invitation from the Technical Society of the Pacific Coast and the Pacific Northwest Society of Engineers, that the members of the Society should be their guests in connection with the Pacific Coast Engineering Congress to be held at Portland, Oregon, as a feature of the Lewis and Clark Exposition. The date set for the congress was June 29th–July 3rd, and an attractive list of papers by members of the home societies, by authors of the United States Reclamation Service and others had been promised, together with excursions of special interest to the Dalles of the Columbia and

the Cascade Locks, and to the dam on the Columbia River, under construction by the Oregon Water Power and Railway Company.

The Secretary reported that by direction of the council the courtesy of this invitation had been acknowledged, and that information could be given any member desiring to participate in it. It was suggested that two members of the society be appointed by the President under the provisions of the by-laws, to represent the Society officially at this congress, and the matter was informally left to the President without resolution, under such provision.

The chair called for a report from the sub-committee representing the Society on the work intrusted with the details of the Engineering Building for the accommodation of the Societies and other scientific organizations. Such report was presented by Mr. Charles Wallace Hunt as follows:

Mr. Hunt.—At the December meeting of the Society a report of progress was made, and I presume the members of the Society are familiar with the statement then made. The committee has taken the view that the building is not alone for the three engineering societies who have its management; hence they have laid their plans so as to make it a convenient building to become a center for leading engineering societies of the United States. Consequently, the building proposed is to be larger than would be required for these three founder societies alone, the idea being to bring other engineering societies into the same building under the term of associate societies. Such societies are to pay a pro rata share of the expense of the maintenance of the building instead of being charged rent, which in other buildings would be done with the idea of making a profit.

The uses of the building, being entirely new, required of the committee a great deal of work to bring the design within the financial limits, and to make it convenient in use. Each society has its own floor of offices arranged to suit the requirements of its special work. In addition, there is on the second floor a main auditorium, and on the floor above smaller meeting rooms, ample in number and convenient in arrangement. The library is laid out on a broad plan, expecting that in time it will be the largest and most complete engineering library in the world.

Twenty-six architects submitted plans for the building. The method for the selection of an architect was very carefully thought

out, and I believe the committee actually secured the best plan and the best architect available at the time.

The main auditorium will seat 1,000 persons. On each side of the room there are corridors where members can converse when not interested in the meeting. There is to be a passageway in the rear of the platform, so that a person wishing to pass from one side of the platform to the other can do so without coming in view of the audience.

The building is to be 100 x 90 feet in plan and 209 feet high. A space is left entirely around the building so that air and light will enter all the rooms. The bids for the construction of the building are to be opened on the 20th of the present month. As soon as the bids are opened and the contract awarded, an engraving of the building will be prepared and sent out to the members of the Society.

The President thereupon made announcement that Mr. Chas. H. Haswell, first Engineer-in-Chief of the United States Navy, and now in active practice of engineering, at the advanced age of ninety-six, had been elected to honorary membership in the Society.

Mr. Haswell was the author of the first engineering pocket-book, and had been connected with all important matters in the early history of the United States Navy. Mr. Suplee in comment spoke of his having been able to secure at a second-hand book-store a copy of the first edition of this pocket-book, published in 1847, and bearing on its title page in Mr. Haswell's handwriting an inscription that it was presented by the author to his friend Captain John Ericsson. It was the purpose of Mr. Suplee to present this souvenir to the archives of the Society on his return to New York.

The Secretary reported that the council at its meeting, June 6th, had directed that the following action be reported to the meeting for publication and record:

Whereas, it has been suggested that in the summer of 1907 a Centennial Anniversary be celebrated to commemorate the culmination of the experiments in

STEAM NAVIGATION

which center around the first voyage of Robert Fulton's *Clermont* in 1807; and,

Whereas, this Society of Mechanical Engineers is proud to recognize the great achievements in mechanical engineering as applied to marine propulsion,

which have marked the period since these modest beginnings of one hundred years ago;

Resolved, that the Council approves the recommendation to hold such celebration of this centenary, and believes that such a commemoration will be both desirable and appropriate.

Mr. Newell Sanders, on the call of the chair, presented on behalf of the industrial interests centering in the city of Chattanooga, Tenn., an invitation to all members to take part in the convention which had been ordered for the spring of 1906 in that city. He spoke of the historic attractions centering around the memories of the contests at Lookout Mountain and Chickamauga, and of the development of the industrial South, which he hoped the visit of the engineers would stimulate and encourage.

No other new business being presented at this point, the meeting took up the discussion of professional papers as follows: "Microstructure and Frictional Characteristics in Bearing Metals," by Melvin Price; "Cast Iron, Crushing Loads and Microstructure," by Wm. J. Keep; "Smoke and Its Abatement," by Chas. H. Benjamin. The participants in debate were Messrs. Balkwill, Barrus, Bement, Blauvelt, Brooks, Freeman, Hawkins, Herschmann, Hutton, Mann, Miner, Moss, Suplee, Swasey, and Taylor.

Third Session. Wednesday Evening, June 7th.

The papers of this evening were as follows: "Can a Steam Turbine be Started in an Emergency Quicker than a Reciprocating Engine of the Same Power," by A. S. Mann; "Note on Efficiency of Steam-Generating Apparatus," by A. Bement; "Performance of a Superheater," by A. Bement; "Counterweights for Large Engines," by D. S. Jacobus; "Steam-Actuated Valve Gear," by Wm. H. Collier. The participants in debate were Messrs. Baker, Ball, Barrus, Blauvelt, Bryant, Bushnell, Cary, Foster, Herschmann, Hutton, Kent, Moss, Rice, Rites, Smith, and Suplee.

Fourth Session. Thursday Morning, June 8th.

This session included the following papers: "Note on Heads of Machine Screws," by H. G. Reist; "Belt Creep," by W. W. Bird; "Function of Laboratory Courses in the Curriculum of

Engineering Schools," by Chas. E. Lucke; "Continuous Measuring and Mixing Machines," by E. N. Trump; "Epochs in Marine Engineering," by George W. Melville. Participants in debate were Messrs. Barrus, Bement, Burlingame, Diemer, Ennis, Hawkins, Hutton, Jacobus, Mayo, Suplee, Whitehead, and Williston.

At the conclusion of the papers the chair called for new business, and Mr. Sanders took occasion to repeat his invitation for the city of Chattanooga, reminding the members that in 1915 would occur the semi-centennial anniversary of the close of the Civil War, and that it was the purpose of that city to arrange for a special exposition in commemoration, and that the Federal Government proposed to erect a memorial arch to rival the arch of triumph at Paris.

The Secretary, on behalf of the committee appointed to record resolutions of thanks for courtesies enjoyed by the visitors, presented as the report of such a body the resolutions of thanks as follow:

The American Society of Mechanical Engineers at the closing session of its fifty-first meeting held in the city of Scranton, desires to place upon its records its sincere thanks for the courtesies in many directions which have been enjoyed during its stay. The Society asks that behind the formal mold of preamble and resolutions, our hosts will see the earnest desire of the members to express what is in their hearts.

Resolved, that the American Society of Mechanical Engineers extends its hearty thanks to the McClave-Brooks Co., and to Mr. William R. McClave, its General Manager, for the invitation to visit the works of that Company and inspect their special appliances adapted for the special conditions of the smaller fuels.

Resolved, that the Society has thoroughly appreciated the privilege of a visit to the Pine Brook Breaker and to study the special problems presented by this important undertaking.

It has been a great pleasure to the ladies of the American Society of Mechanical Engineers that an opportunity should have been given to visit the Scranton Lace Curtain Co., and to study in detail the interesting and artistic products of that Company.

Resolved, that the thanks of the Society be and hereby are extended to the Klotz Throwing Mill for the chance to inspect the interesting process for the making of silk yarn for ribbon and other purposes. The Power plant of this establishment has been of special interest to the visitors.

Among the members of the American Society of Mechanical Engineers are many employers in widely differing lines. Probably in no other city in the country could so many men find an institution having such broad and pervasive interest as the visitors have found in the International Correspondence Schools. The visitors ask that the Trustees and management of this Institution, and that Messrs. R. J. and T. J. Foster in particular, together with their gifted

and energetic staff of officers and co-workers, will receive the hearty thanks of the members of the Society for the opportunity to visit their surprising, complete and well-organized institution. They would ask that they may record their appreciative pleasure in the system by which their visit was organized and conducted. The members believe that to a very great many persons the opportunity offered by the Correspondence School is a door to hope and opportunity, whose influence will last through many years and be of unpredictable good to an uncountable number.

Resolved, that the American Society of Mechanical Engineers extends its hearty thanks to the management and officers of the Allis-Chalmers Works in Scranton for the courtesy of the invitation to visit the Scranton plant, and for that care taken that the visitors should see all points of mechanical interest at that point. It does not often happen that under one roof the mechanical visitors can study the transition from the slow-speed belt-driven machine tool to the high-speed electrically driven type, and the opportunity for this examination was much appreciated.

Resolved, that the American Society of Mechanical Engineers desires to place on record its sincere appreciation of the courtesy enjoyed at the hands of the D. L. & W. R. R., and its officers and management, for the invitation and privilege to visit their plant at Keyser Valley. As an example of advanced modern practice in arrangement, construction and interior detail, the Society is pleased to have an opportunity to express its pleasure in their study. The Society would include, besides the Car shops, the Boiler and Power plant, and the Water hoist at Hampton.

Among the pleasantest memories of the attractive surroundings of Scranton will be the recollection of the visitors of the beauty of the line known as the Laurel Line of the L. & W. V. R. R. Co. The members ask that the management of that Company will receive the sincere thanks of the Society for the courtesy of the special train which took them for a visit to the Power house, and for the run over the viaducts and other interesting mechanical details of the alignment.

Resolved, that the thanks of the Society be extended to the Wyoming Valley Traction Co. for the courtesy of special cars in Wilkes-Barre, and for the privilege and opportunity of visiting the Vulcan Iron Works of South Wilkes-Barre, the Sheldon Axle Works in North Wilkes-Barre, and for the invitation from the Hazard Mfg. Co., which, the time of the visitors available in the press of other invitations, it has been impossible to accept.

The Society requests his Honor, Mr. Kirkendall, Mayor of Wilkes-Barre, to accept the sincere appreciation of the visitors for the interest which he has evinced in their visit, and for the courtesies enjoyed at his end.

Resolved, that the courtesy of the Lehigh & Wilkes-Barre Coal Co. is most heartily appreciated, and the tender of their invitation to visit their No. 5 Colliery in Wilkes-Barre.

Resolved, that the American Society of Mechanical Engineers extends its hearty thanks to the Scranton Engineers Club, Mr. W. M. Marple, Past President, and Mr. E. M. Zehnder, President, for the courtesy of their invitation, by which members of the Society have been made guests of the Club during their stay in Scranton, and for the effective co-operation of the Club and its members

in carrying out the purposes of the meeting. The members have only to regret that the unusual interest attaching to the program of excursions has prevented the members from availing themselves of the proffered courtesy.

The members and officers of the Society tender their hearty thanks to the Scranton Club for the visitors' privilege offered by the President of the Club during the stay of the Engineers in Scranton. The days have been filled so full that opportunities for social relaxation have been scarcely possible.

In addition to the attractive room furnished to the Society under a business arrangement with the Y. M. C. A. of Scranton, that body has, through its Secretary, put at the service of the visiting engineers the privileges of their building as enjoyed by members of the Association. For the tender of available rooms, for the use of the gymnasium, swimming-pool and other athletic facilities, the visitors extend their sincere thanks.

The visiting ladies accompanying the members of the American Society of Mechanical Engineers have been greatly favored by the interested provisions which have been made for their pleasure and enjoyment. As the gentlemen of the Society have to be their mouthpiece in official action, the ladies ask that through the vote of the members their sincere thanks may be conveyed to Mrs. Chas. S. Weston and her friends for the automobile ride around Lake Scranton and up Mt. Anonymous. They ask also that by this action the thanks of the ladies and the Society may also reach Mrs. L. A. Watres, to whose courtesy is due a most attractive experience during their stay.

The ladies ask also that by this official vote of the Society a record may be made of their appreciation to Mrs. J. Benjamin Dimmick for the graceful hospitality of which she was the channel at her residence on Wednesday afternoon. It is one of the pleasantest elements of the visitors' experience to be taken into the charming atmosphere of an American home when the Society is a guest of the professional interests of an industrial city.

Resolved, that the thanks of the Society are cordially extended to the Board of Governors of the Bicycle Club of Scranton for the courtesy by which the Club House and its facilities have been put at the service of the Committee and the members for the most enjoyable reception on Thursday evening. The privilege of meeting the resident representatives of the city and the pleasure of the brilliant evening have been much appreciated.

Resolved, that the Scranton Street Railway Company and Mr. Wildt, its Mechanical Engineer, be asked to accept the sincere thanks of the Society for the courtesy of free special cars and other attentions in respect to transportation for the members of the Society, and for the invitations to visit the Power House of the Company.

Finally, the Society seeks by its closing resolution to convey to the members of the local committee, Mr. W. F. Mattes, Chairman, and to his associates, chairmen of the effective Sub-Committees, their sincere appreciation of the efforts of the Committee in arranging the program in securing the privilege of visit to works, mines, factories and institutions, and for the assiduous care by which the individual desires of the visiting members have been consulted and their convenience served. Where a successful undertaking has been the result of effective co-operation of many minds and hands, it is difficult to signalize individuals without danger of mistake. It is particularly, however, to Mr. A. B. Dunning, Chairman of the Excursion Committee, and his assistants, that

the smooth working of our several visits during the Scranton stay has been manifestly due. To him, to his co-workers, and to all residents of Scranton who have co-operated with them, the Society is greatly indebted. The meeting asks that Mr. Mattes and the local Committee will convey to the management of the Consolidated Telephone Co. and the Pennsylvania Telephone Co., the appreciation of the courtesy by which their lines have been brought into the headquarters of the Society in Scranton for the use of the officers and members during their stay. The members ask that any other courtesies, which, by reason of their inconspicuous character, the Society has not been able to emphasize, the local Committee will take upon itself to acknowledge, in the name of the Society.

The visitors leave Scranton with regret that they must, in leaving, have missed so much of what the hospitality of our hosts would fain have placed in our way. They can only express the hope that when the meeting shall fall in their own city, the Engineers of Scranton will make it a point to attend such conventions in large numbers, that an opportunity may be given for the exercise of reciprocal courtesies.

The resolutions above quoted report in some detail the provisions for entertainment of the visitors in the matter of excursions.

On Tuesday afternoon the visitors were escorted by the Excursion Committee to the plant of the McClave-Brooks Company and the Pine Brook Breaker.

On Wednesday the excursions included the International Correspondence Schools.

On Friday the excursion was allotted for an all-day visit over the Lackawanna and Wyoming Valley third-rail tramway to Wilkes-Barre and included a visit to the power house.

At Wilkes-Barre the members were entertained at luncheon by the local residents and welcomed by the Mayor of the city, the Hon. Mr. Kirkendall.

After luncheon the excursion included the Vulcan Iron Works, the No. 5 Colliery of the Lehigh and Wilkes-Barre Coal Company, the Hazard Manufacturing Company, and the Sheldon Axle Works.

For the ladies of the party a most attractive special program was arranged on Wednesday, by which they visited under escort the Scranton Lace Curtain Company and the Kolt Throwing Mill, and in the afternoon a most attractive reception was tendered at her residence by Mrs. J. Benjamin Dimmick.

On Thursday morning the ladies were the guests of the local ladies for a drive in automobiles around Lake Scranton and Mt. Anonymous, with the opportunity of seeing the attractive views on the road.

On Thursday evening the members and their ladies were guests of the Engineers' Club, the Board of Trade and the citizens of Scranton, in the attractive club house of the Scranton Bicycle Club. The officers and the lady patronesses received the visitors, and music and dancing continued until a late hour.

No. 1076.*

EPOCHS IN MARINE ENGINEERING.

BY GEORGE W. MELVILLE, REAR-ADMIRAL, U.S.N., RETIRED, PHILADELPHIA, PA.
(Member of the Society.)

1. To attempt to cover the history of marine engineering in a lecture which ought not to last much longer than an hour, would result in little more than a mere chronology and could not possibly be interesting. It seems much better, therefore, to give some consideration to the various periods or epochs in the history considered with reference to the special inventions or changes which have characterized them. Many of these changes are clear examples of evolution and others are instances of the adaptation of land practice to marine use, but they all serve to show the constant progress which has taken place.

The Paddle Wheel.

2. The first marine engineering in the modern sense was the adaptation of the steam engine as already in familiar use on shore to a modification of the centuries old method of mercantile propulsion, the oar. I believe some attempts were actually made to adapt the steam engine to a series of oars, which would have meant something like a mechanical trireme; but of course the trained mechanical sense soon saw that the collection of the oars in a revolving wheel was the correct solution. As oars had been used on both sides, so it was natural at first that the paddle wheels should be on both sides, a center wheel was also tried, but it is interesting to remark that practically about the same time that the sidewheels were used on the seaboard, the first marine engine was the shore engine modified to suit the circumstances, and thus

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on the seaboard the engine was designed and worked with what we now consider an exceedingly low pressure. On the western rivers where the change has been made in the location of the wheel, there was also the additional change of dispensing with the condenser and using very much higher pressures. It was doubtless due to this fact—that the first non-condensing engines really carried a very high pressure—that the term "high pressure" in the early days meant non-condensing. The reason for the difference is of course very clear; the western rivers are very shallow and it was necessary to make the machinery as light as possible; on the seaboard and the rivers of that section there was deep water and the vessels had displacement enough to permit of heavy machinery.

3. Ordinarily the history of this olden time could have only an antiquarian interest for us, but we are unusually fortunate in still having with us in the active practice of his profession an engineer who saw the first commerciallly successful steamboat, the "Clermont," so that through our "grand old man of engineering," Mr. Charles H. Haswell, we still have a living connection with that earlier time. One of my former assistants some years ago contributed a series of articles to one of the engineering magazines and has given some data as to the performance of the FULTON, the first American steam man-of-war, whose machinery was designed by Mr. Haswell, who was also her first chief engineer. An extract from the "Fulton's" steam log for January, 1838, shows that the maximum steam pressure was eleven pounds, the vacuum twenty-four inches and the maximum revolutions per minute eighteen.

The Screw Propeller.

4. As time passed on and experience was gained, there was naturally an improvement in workmanship and design and a moderate increase in steam pressure, but about 1836 the screw propeller was brought forward for driving large vessels. This was not the first application of the screw propeller, which had been used successfully on a small steamer or launch about 68 feet long as early as 1804 by Col. John Stevens, the grandfather of Col. Edwin A. Stevens, who is now so active in marine work, and the father of Robert L. Stevens, who was the most famous of the name for his work in connection with marine engineering. It is to be noted that this date is prior to that of the building of the "Fulton,"

but naturally, in the first steam war vessel, it was not considered desirable to do anything of an experimental nature, a condition which was obtained to some extent ever since and probably always will. It was about ten years later that the propeller began to come into general use and entirely displaced the paddle wheel for ocean-going steamers. The reason for this change is easy to see on a little consideration. On a long ocean voyage the change in displacement is due almost entirely to the consumption of fuel. In the case of the propeller this makes practically no difference in its immersion or in its efficiency, while in the case of the paddle wheel the immersion of the floats would be changed, with a diminution of efficiency. Further than this, the paddle box offered very great resistance to strong head winds and also brought severe stresses on the ship, due to rolling in heavy seas, the propeller not being affected by either of these last two causes. To-day for work in deep water, the screw, of course, is the only propeller; but for river work, particularly in shallow rivers, the paddle wheel is still used. Efforts have been made, and some of them exceedingly ingenious, to adapt the propeller to use in shallow water, and a certain amount of success has attended the efforts of such brilliant engineers as Yarrow and Thornycroft, not to mention our own Mr. Charles Ward, who has done some work in this connection. The fact remains, however, that damage by sand bars, snags, etc., may easily render screw machinery inoperative, while as expressed by an engineer who had designed many western river boats, "any wood butcher can repair a stern wheel."

5. Probably fully as much has been done to improve the design of propellers, as time has gone on, as any other part of marine machinery. In the early days the rules for propeller design were exceedingly crude, but with the slow engine speeds which then obtained the effects were not noticeable. As engine speeds increased, however, it was seen that these old rules were utterly inadmissible. There is no good excuse, however, for progress having been so long delayed, for the designs still remained too crude even after Isherwood's famous Mare Island experiments in 1868. Probably one of the great troubles with screw propeller design at the beginning was the mistake made in considering the action of the screw as analogous to that of a bolt working in a nut, from which it was inferred that the smaller the slip the greater the efficiency. As a matter of fact, a screw propeller is really a pump for driving a mass of water astern, the reaction from which drives

the vessel ahead. When this had once been realized, it was seen that there must be a certain amount of slip and that under proper conditions there could be a relatively large slip and still high efficiency.

6. Multiple screws were used as early as our Civil War on some vessels known as "tin-clads" on the Mississippi, their adoption being necessitated by the shallow draught. Twin screws were first used in war vessels where the necessity for keeping the machinery below the deck would not allow of all the power being conveniently used on a single shaft, but the great advantage they possess of security against total disablement and for maneuvering soon made them the rule for all naval vessels large enough to admit of them. They were much longer in coming in the merchant service where the limitations on naval machinery do not obtain; but since the era of the very large trans-atlantic steamers beginning with the "Paris" and "New York" and the "Teutonic" and "Majestic"—all very large vessels have been built with twin screws.

7. In the navies of France and Germany, the triple screws have been used to a considerable extent and I used them myself in our own fast cruisers "Columbia" and "Minneapolis." My own belief is that they have decided merit for vessels using about 10,000 horse-power. This view, however, was not shared by my Board colleagues in the Navy Department, so that their marked success in the two vessels named was not allowed to be repeated in later designs.

8. With the advent of the steam turbine as a prime mover in ships multiple screws have again come to the front, this time on account of the extremely high speed of rotation of the shafts. The "Turbinia," the first vessel of this kind, had three shafts with three propellers on each; and the destroyers "Viper" and "Cobra" had four shafts with two propellers on each. The merchant vessels, beginning with the "King Edward," have been fitted with three shafts.

9. Before leaving the propeller, I may mention, in connection with the improvements in its design, the care that is now taken to avoid needless friction by making the hub spherical with a conical tail-piece, and by putting covering plates over the bolts, securing the plates to the hub so as to continue the outline of the sphere or conoid.

The Surface Condenser.

10. Up to about 1860 the jet condenser was the one usually employed on board ship, which meant, of course, that the boilers were constantly fed with salt water; and this, in turn, meant the deposition of great quantities of sulphate of lime scale on the heating surface. With the low pressures then prevalent this did not greatly affect the economy of the boilers, except that, as "blowing off" to keep the density of the water down was a continuous practice, there was a certain loss of heat, and of course there was the necessity for frequent scaling of the heating surfaces. However, they were effectually protected against corrosion. About 1860 the use of surface condensation became general, and as this greatly reduced the amount of scale formed, it was practicable and safe to increase steam pressures, which accordingly resulted with a consequent reduction in the weight of machinery per unit of power.

11. An accompaniment of the introduction of surface condensation, which was at first supposed to be a result of it but which as a matter of fact was not, was a tremendous increase in the corrosion of the boilers and shortening of their life. This was especially noticeable in the tubes which, as the thinnest part of the boilers, give out first. All sorts of theories were advanced to account for it, some of which we can now see to have been utterly ridiculous. Probably one of the most fanciful was that which regarded the boiler and condenser as forming a gigantic galvanic battery, the copper condenser tubes forming one pole and the boiler the other. The real facts were developed as a result of the investigation by the Admiralty Committee on boilers in the 70's, which showed that boiler corrosion was simply rusting and had been due to gross but unintentional neglect.

It had been a very common practice, particularly in naval boilers, when they were not in use, to blow out the water and take off the manhole plates "to let them air." It was this "airing" which caused the corrosion. Now when boilers are laid up, they are filled with water which is made slightly alkaline, and this effectually prevents corrosion.

The Cylindrical Boiler.

12. The early boilers in sea-going vessels were of what has been called the "box" type; that is, the boiler was a cubical box with a thin shell, the real strength being given by braces running

in three directions. When surface condensation had made higher pressures possible, it was soon found that the multiplicity of braces, as pressures were increased, made an impossible condition of affairs, and this led to the design of the cylindrical boiler whose shell was self-bracing and left the only braces, those needed for the heads and flat surfaces. This boiler so thoroughly met the conditions arising that it has remained the favorite even up to the present day. At one time an effort was made to save room by making the boiler elliptical, but this was soon found to be unsatisfactory and impracticable and was abandoned after only a few examples.

13. The earliest cylindrical boilers were single-ended with two furnaces, but with the advent of reliable mild steel the diameters were increased and the boiler was made double ended, with upper ends rounded to save bracing, so that the largest cylindrical boilers to-day have as many as eight furnaces, four in each end in pairs; that is, the two furnaces at each end on the same side of a vertical diameter have a common combustion chamber. The saving in weight due to the double ended boiler is evident at once and also the reduction in the feeding apparatus required.

14. Notwithstanding the advent of the water tube boiler, which will be mentioned further on, and its practical preëmption of naval practice, the cylindrical boiler still remains the favorite for the merchant service and has been used for pressures as high as 220 pounds, even in the largest sizes on such vessels as the "Kaiser Wilhelm." The highest recorded pressure is 255 pounds on the "Inchdune."

The Compound Engine.

15. From a very early period steam has been used expansively in marine engines, and indeed sometimes to a ridiculous extent. Some engineers as late as the Civil War hardly seemed to realize that there was any limit to expansion, although Isherwood's famous experiments on the "Michigan" in 1861 had demonstrated conclusively that, with low pressures, only a very moderate expansion is permissible, beyond which any further expansion is attended by an economic loss. As pressures increased it was natural and correct that a higher range of expansion should be used, and this made practicable the compound engine, where the expansion occurs in two stages, the high pressure steam from the boiler being limited to a small cylinder from which, in turn, the steam of

lower pressure is exhausted to a larger cylinder. As you all know, the compound engine was invented almost as early as Watt's separate condenser, Hornblower's patent dating back to 1771, and Wolff's patent for a two-cylinder engine dating back to 1804. With the low pressures prevalent at that time the compound engine was actually at a disadvantage compared with the simple one. When pressures had gotten up to about 60 pounds, however, the compound engine began to assert itself, the pioneer in that respect being John Elder, of the firm of Randolph & Elder, which is now known as the Fairfield Engine Works. It is interesting to note that the Allan Line of steamers, which is now the pioneer in introducing the steam turbine for an ocean-going steamer, made the last scientific stand against the compound engine, going so far as to take duplicate vessels, and engine one with compound and the other with simple engines of the same power. The actual experience with these two vessels where the simple engine with the high ratio of expansion was constantly in trouble from breaking down, was a convincing proof that high ratios of expansion in a single cylinder was impracticable.

16. With the improved workmanship which had come by this time and with the improvement in materials, to which we shall refer in a moment, which came later, the compound engine advanced to a high state of perfection and for large powers the three cylinder type, with one high pressure and two low pressure cylinders, became a favorite for all large vessels. These engines were probably as fine specimens of marine engine designing as have ever been seen, and included some exceedingly ingenious valve gears designed to secure variable expansion and an equalization of work among the various cylinders. As we shall see later, the further advance relegated these beautiful mechanisms to the engineering museum.

The Advent of Mild Steel.

17. It is probably difficult for the young men in our technical schools of to-day who are familiar almost entirely with mild steel and very little with wrought iron, to realize what a change came in engineering when the production of mild steel became a commercially reliable matter. When we look back at the way in which some of the vital elements of a big marine engine was made, we are almost inclined to wonder that the material was reliable at all. The difference between a large wrought iron shaft such as

old Hughey Dougherty used to make at the Morgan Iron Works, and one of the mild steel shafts made at Bethlehem, is as great as could be imagined. Nearly the same is true of boiler plates. The young engineer of to-day would hardly know what was meant by a lamination or a "cold shut." The very method of manufacture made it necessary to use a large factor of safety in designing, with the result that the working stresses permissible were very low and the weight of machinery inordinately high. With the advent of mild steel and the introduction of careful and systematic testing, the designer had a material on which he could place absolute reliance so that the factor of safety could be greatly reduced. As a matter of fact the factor of safety has been reduced from 8 or 10 to 5, and sometimes as low as 4.5.

18. In looking back over my own experience, I do not see how we could possibly have built engines of the size and power now common with wrought iron for piston and connecting rods and shafting, and it is, of course, absolutely certain that we could not have built cylindrical boilers of to-day. The change began in the later 70's and had become almost complete by the middle of the 80's.

19. We must not fail to notice in the change to steel the use of steel castings, which have displaced cast iron in many places with attendant great reduction in weight. The first use of steel castings was attended with considerable annoyance because it was unfortunately assumed, perhaps naturally, that, barring the much greater shrinkage, it could be treated very much the same as cast iron, and it was consequently asserted with confidence that anything which had been made in cast iron could be made in cast steel. That is doubtless true to-day, but it certainly was not true ten or fifteen years ago, as I know to my personal sorrow, because designs which would have been considered simple in cast iron had to be entirely changed to meet the conditions then existing for steel castings.

20. It may be well to mention in this connection that about the same time that steel came into use displacing wrought iron, white metal for bearings and the stronger bronzes also came into use, thus giving the designer much better materials to work with and again reducing weights.

21. We may also mention here the gradual displacement of copper for steam piping by steel pipes, owing to the fact that for the high pressures common at present copper pipes would have

to be very thick, making it difficult to secure a sound joint, and also to the serious diminution of the strength of copper by the high temperature.

Forced Draft.

22. Forced draft dates back of course to Stevens' Rocket, and its first use for marine purposes was by Mr. Robert L. Stevens on the Hudson River steamers in our own country prior to the Civil War. During that war Mr. Isherwood built a number of gunboats which used forced draft, but it had fallen into disuse until about 1882 for naval vessels, when it was introduced into the English navy and still later was applied in the merchant service.

23. In naval machinery forced draft has been of the greatest possible importance, because it has reduced boiler weights probably almost one-half. In the navy the natural limitations as to space and weight prevent the use of forced draft with very much economy of fuel. It is obvious that if the rate of combustion is increased from 15 pounds of coal per square foot of grate to 40 pounds, there ought to be an attendant increase of heating surface. In the merchant service, or at least in certain classes of vessels in that service, it is possible to do this, and in one of my annual reports I made a comparison between the boilers of a merchant vessel called the "Iona" and those of the "Baltimore." In the "Iona" there were 75 feet of heating surface for 1 grate, while in the "Baltimore" the ratio was about 30 to 1; but had the "Baltimore's" boilers been designed with any such ratio, their weight would have been almost double the weight of all the machinery of that vessel as actually built.

24. Mr. James Howden has made a speciality of forced draft under economical conditions, heating the air before admission to the ash-pit; and his system is now in use on most of the large steamers, the aggregate horse-power running up, I believe, to over a million.

High Engine Speed.

25. About the same time as the reintroduction of forced draft in naval vessels, the improved materials and workmanship made it possible to get higher rotational speeds, and, as remarked earlier, the true conditions of propeller design being understood, there was a marked increase in the speed of rotation of the en-

gines. Naval vessels, from the necessary limitation of keeping the vital parts of the machinery protected as far as possible, have always run faster than the engines of the merchant service, although this did not always mean that their piston speed was greater. The mistake is sometimes made of attributing lighter machinery to higher piston speeds, but unless this is accomplished by increasing the number of revolutions it will not produce that effect. In the early days, 60 or 70 revolutions per minute for what was then considered a large engine of 4,000 or 5,000 horse-power, was about the limit, but in engines of as much as 8,000 horse-power for a single set one finds the revolutions to-day as high as 130. Of course it is not practicable to show to just what extent any one line of progress has reduced weights by comparing the machinery at periods wide apart, because the increase of steam pressure, increased rotational speeds, improved materials and better designing have all gone along together—but it is interesting to note that in the "Warrior" of 1861, with 22 pounds boiler pressure and 54 revolutions, the horse-power per ton of machinery was 6, while in the "Minneapolis" of 1891, with 165 pounds pressure and 133 revolutions, the horse-power per ton is 10.9. From a simple mechanical standpoint, contributing agencies to the high speeds are the much more perfect journals of the steel shafts and the superior white metals used for bearings, and the rigidity of the steel hull of the ship, as compared with the older conditions. The best makers now grind their bearings true to a mandrel which represents perfect alignment. In the old days all main shaft bearings were hollow for water circulation, which was generally needed, and there was usually provision for a spray of water on the crank pins. In the modern engines which have been well built and are carefully adjusted, there is no necessity for water even at very high speeds under full power.

The Multiple Expansion Engine.

26. The change from the simple to the compound engine involved a certain amount of difficulty and opposition, but the lesson was then learned pretty thoroughly that the amount of expansion in a single cylinder was moderate. Consequently as steam pressures rose the leaders of the profession became convinced that to secure adequate economy a further stage of expansion was necessary, and this brought about the triple expan-

sion engine, the credit for which is deservedly given to Dr. A. C. Kirk, of the firm of R. Napier & Sons, of Glasgow, who first successfully used the triple expansion engine on the steamer "Aberdeen." The adoption of the triple expansion engine was almost immediate, and, after the success of the "Aberdeen" was demonstrated, all new engines of any size were built on that principle. It seemed a logical extension of this idea that with still further increase of pressure there should be the quadruple expansion engine, and a number of these have been built; but the advantage as compared with the triple expansion engine up to the point beyond which pressures have thus far not gone, does not seem to be clearly demonstrated, and a great many designers are adhering to the triple expansion engine with four cylinders, one high, one intermediate, and two low pressures.

Water Tube Boilers.

27. Like so many other details not only in marine engineering but in other lines of work, features which are introduced in a practical way in recent times are found to have a comparatively ancient origin. This is true of the water tube boiler, which in its recent use dates from about 1880. The excavations at Pompeii have shown small boilers almost identical in construction with some of the best of our water tube boilers, although they were doubtless only used for a circulation of hot water.

28. The object of the water tube boiler is to reduce weights, give greater safety against explosion, greater rapidity of raising steam, and an increase of economy in the generation of steam. The various makes of water tube boilers are too numerous to mention, but they divide themselves into two broad, general classes; those with straight tubes of large diameter, say four inches; and those with curved tubes of small diameter, from an inch to an inch and a half. Probably no single boiler possesses all the merits which a perfect water tube boiler should have, and in nearly every case the attempt to secure certain advantages brings attendant disadvantages, and vice versa. The large straight tube boilers are not so light as the ones with small tubes; and it is more difficult to secure adequate economy, which is dependent largely upon skillful baffling. They do not permit of such rapid raising of steam from cold water as the smaller tube boilers. Because, like the Scotch boiler, they carry a large reserve supply of water

in the boiler after the manner of the Babcock & Wilcox boiler. On the other hand, they permit the replacement of a defective tube and of the cleaning of a tube much more readily than the tubes which are bent. Likewise it is only necessary to carry one size of spare tubes, while the bent tube boilers require several sizes and shapes.

29. As far as safety against explosion is concerned, there can be no doubt that there is less danger of an actual disaster affecting the whole ship—although the worst accident which we ever had with a boiler in my naval experience was in connection with a water tube boiler on a torpedo boat, where all the crew of a fire room were scalded to death. Nevertheless, the boiler itself did not explode and was quite easily repaired. On the other hand, a few years before this a locomotive boiler on a torpedo boat in Germany exploded through the collapse of the crown sheet due to low water, and not only killed all the people in the fire room, but tore up the decks and utterly ruined the boiler itself. In this connection it is a cause of sincere congratulation that since the explosion of the "Thunderer's" boilers in the English navy many years ago, there is, I believe, no record of the explosion of a large, well built marine boiler. For naval purposes where weight is such a great consideration, the water tube boiler commended itself at once, and it has now become the established practice in all navies to use only water tube boilers in new ships. Our English friends had some trouble with the Belleville boiler, but this seems to have been due to some extent to lack of familiarity with it, and other legitimate reasons. In the merchant service where weight is not so precious, the water tube boiler has not thus far so thoroughly commended itself to designers; and, as remarked earlier, all of the latest large vessels are still using cylindrical boilers. Some of the reasons for the hesitation to adopt water tube boilers are that, of necessity, an installation of large power means a very large number of boilers, because the water tube boiler does not admit of single units of great power comparable with the large double-ended cylindrical boilers. This means an immensely greater complication in the way of piping, valves, fittings, feed pumps, etc. Moreover, owing to the small amount of contained water, which is very desirable in so far as weight is concerned, the water tube boiler is very sensitive as regards steam pressure and water level, requiring very much more care and attention than the cylindrical with its immense amount of con-

tained water. It seems to me not impracticable that the able men who are engaged in the study of this question will finally succeed in developing a type of water tube boiler which will commend itself for use in the merchant service as well as in the navy. Some of the boilers fitted in the United States naval ships had but six minutes of water endurance after the pumps stopped working, while one of those, the Babcock & Wilcox, adapted to our naval use has as much as twenty-five minutes endurance, which is a close approximation to the Scotch boiler.

Auxiliaries.

30. In the early steamers, almost the only independent steam auxiliary was a single pump which could be used for feeding the boilers while under banked fires or with the engine stopped, and for pumping the bilge. The other pumps were attached to the main engine. Such things as steam capstans, and winches, steam steering gear, distilling apparatus, evaporators, forced draft blowers and electric light engines, were not dreamed of. As time went on and the size of vessels increased, steam capstans and winches and steam steering engines came in. Then it began to be found desirable, particularly for naval engines, to remove all the pumps from the main engine, leaving it nothing to do but turn the propeller, and this brought about independent air and circulating pumps and feed pumps. Further progress introduced the distiller and evaporator, the forced draft blowers and the electric light engine.

31. Most of these auxiliaries, from the nature of the case, are driven by simple engines, the pumps usually being for very slow piston speed and without expansion. The result is that the economy of these auxiliaries is naturally very low, and for some years past it has been the aim of designers to do something to either make the auxiliaries themselves more economical or at least utilize the heat in the exhaust steam. In some cases it has been arranged to have the auxiliaries in the engine room take their steam from the first receiver and exhaust into the second, thus, in effect, making all their cylinders part of the intermediate cylinder, as far as the steam cycle is concerned, with its attendant economy. I remember one of my former associates telling of how he had actually tried this on his ship with a saving of some six tons of coal a day, for machinery which was then work-

ing at about 8,000 horse-power. In the case of the feed pumps, arrangements have been made to turn the exhaust steam into the suction pipe, thereby having this heat carried back into the boiler, but this has not been done to any very great extent. Another plan has been to turn the exhaust from all the auxiliaries into a feed water heater through which the main feed to the boilers would go, and this has been attended by very good results.

32. The question has been raised repeatedly by electrical engineers that it would be a good plan to drive the auxiliaries by electric motors, on account of their very high efficiency even at fractional loads. For the engine room auxiliaries and the boiler feed pumps it seems to me that this is unreasonable, for it means increased complication and certainly an increase in weight; and it has never been shown to my satisfaction that there would be any material increase in economy, owing to the fact that the dynamos as usually supplied on board ship are not large enough to have very economical engines, and from the fact that there are so many machines each with an efficiency less than unity in the circuit between the boiler and the final pump.

33. There are some auxiliaries on board ship, however, where it would seem motors could be used to advantage, notably for driving the forced draft blowers. From the necessities of the case there are usually stowed in hot and rather inaccessible places and it is difficult to keep the engines in good adjustment. It hardly seems to me, however, that direct current motors are well adapted to this service, as they would have to be of the enclosed or dust-proof type, which reduces their heat carrying capacity, and as they have to go in places where the heat is already very great. If alternating current machinery were installed, it seems to me that the induction motor with its extreme simplicity and ability to withstand very hard usage would be especially adapted to this work.

34. It is doubtless known to all of you that in the modern ships the turret turning is done by electric motors. While this is an engineering matter, on board ship it comes within the perview of the ordnance officer, and therefore I have not dwelt upon it. The anchor capstan and steering engine should be electrically driven, but because of peculiar conditions existing in the navy this has not been done.

The Steam Turbine.

35. The latest note of progress in marine engineering seems to be the advent of the steam turbine, which for some purposes has passed the experimental stage, and has given great satisfaction. As already mentioned, a number of small vessels of the torpedo boat class have been built with steam turbines, and this has been followed up by their use in a number of excursion steamers and cross channel steamers between England and France and England and Ireland. The Allan line of steamers have also decided to equip a large new steamer with turbines, and, as we all know from the technical press, the Cunard Company have had a very able committee considering the question of their adoption for the two new flyers which that company is to build—as a result of which they have decided to use turbines in them.

36. The turbine has had an extended use in the last four or five years on shore for driving electric generators, and this has been so satisfactory that the pioneer work of the Westinghouse Company in this country has been followed by the General Electric Company, the Allis-Chalmers Company and a number of other engine builders, who seem to have reached the conclusion that for large powers at least the turbine is quite sure to supersede the reciprocating engine.

37. For constant speed at its rated load the turbine is very economical, comparing in this respect with the most economical reciprocating engines, and its economy does not fall off so rapidly with the decrease of load at constant speed as is the case with the reciprocating engine.

38. For marine purposes two questions have bothered those who were seeking information, one, the question of reversal, and the other, that of economy, where not only the power but the speed is reduced, as is, of course, necessary in the propulsion of vessels. With respect to the former, different methods have been suggested and tried in different cases. In some instances Mr. Parsons has used separate reciprocating engines which are normally idle. In others, and this appears to be in his latest practice, there are reversing turbines inside the exhaust passage of the ahead turbines, so that while the ship is going ahead these turbines revolve idly, after the manner of a fly-wheel, in an excellent vacuum. When the ship is to be reversed, steam is admitted direct to these turbines and secures the reversal.

39. With respect to economy, there is, of course, a marked falling off from that at full power, but not more so than in the case of reciprocating engines. It seems hopeless to expect that any machine will work with the same economy at one-tenth power that it does at full power, and it would be unreasonable to expect the turbine to do this when the reciprocating engine does not.

40. The advantages of the turbine are, the entire absence of reciprocating parts, of bearings to be adjusted and the extreme simplicity of operation, together with the great reduction in weight due to the very high speed. Added to this is the entire freedom from danger due to priming of the boilers, the only result being a slowing down of the turbine and reduction of economy. There is also freedom from lubricating oil getting into the boilers.

41. Propeller design with the turbine is more difficult than with the reciprocating engine, because the conditions are entirely different from those which have hitherto obtained, and there is so little experience with propellers running at speeds of over 1,000 revolutions a minute in the case of small ships, or at 500 to 750 in the case of large ones. The Cunarders propellers, it is understood, are to be limited to 180 revolutions per minute. For it must be remarked that in spite of the fact that we now have very clear and logical rules for the design of propellers under existing circumstances logically worked out, nevertheless these rules and formulæ came after the experience rather than before. This matter, however, can undoubtedly be cared for, and when more experience has been gained the design of propellers will be as easy for existing conditions.

42. For naval vessels from a military standpoint the turbine has a great deal to commend it, inasmuch as the machinery will stow very well in the ship and be out of harm's way, the propellers are so small and so well immersed that there is no chance for racing even in the heaviest seas, and all questions of vibration are eliminated. As already mentioned, the saving in weight is also a matter of decided value, if it can be done.

Conclusion.

43. We have now considered very hastily the important epochs in the history of marine engineering, and it will be seen that they indicate steady progress along certain lines. There has been a steady increase of steam pressure, increase of rotational speed

and diminution of weight, accompanied by increased economy in the making and using of steam. If time had permitted it would have been very interesting to compare such a vessel as the "Great Eastern," which for so long a time was the criterion of immense size, with the "Celtic" and "Cedric," which are even larger than she was. The "Great Eastern" was simply about thirty years ahead of her time. She was a remarkable production and a great credit to her designers in every way, but she was a commercial failure, because marine engine manufacture had not yet reached the point where such a huge vessel could be operated profitably. She carried more than twice as much coal as one of these present day vessels, which easily makes the same speed with machinery weighing probably not more than one-third of what her's did.

44. The economy of the marine engine, or of the turbine if it displaces the engine, has reached a point which does not leave much room for improvement in materials and workmanship. Yet it will be unreasonable to believe that we have reached finality, for it is likely that there have been numerous periods in the past when the designers of those days could not see what the next step in advance would be, and so far as their knowledge went their design was nearly perfect. Of course there was certainly plenty of margin for increased economy with them from which we are barred, but I have no doubt that if I could live to be as old as Mr. Haswell I should see some decided improvements in the course of the next thirty years.

DISCUSSION.

Mr. John T. Hawkins.—While denominating what follows as a discussion of the above-named paper, it is really more supplementary to than a discussion of it. Having been contemporaneous with Admiral Melville in the U. S. Navy up to the date of resignation in 1869, and of the same class or date as the Admiral in entering the service—July 29th, 1861—the writer has naturally been in touch with much of what he has so felicitously given us in his paper; and particularly in connection with that epochal feature in marine engineering, the advances made in our time which involve the question of the use of steam expansively.

With reference, then, more particularly to paragraph 15 of the paper: "The Compound Engine," it is hoped to add something of interest in that connection, as prompted by some experi-

ences during the Civil War, in illustration of what has really been the most far reaching and epoch making feature in marine engineering advance during the past half century.

It is a far cry from the days of the side-wheel river steamer with its Stevens cut-off, admitting steam of about 25 pounds boiler pressure for more than half the stroke, or the side-wheelers of the Cunard and now long defunct Collins lines of ocean steamers, with their cumbrous side lever engines, or similar types in the navy such as the old frigate "Mississippi," working under similar conditions, to the present ocean greyhounds or Admiral Melville's battleships and cruisers, expanding steam of 250 pounds serially in multiple cylinders through ten times greater range; and as the passage from one to the other has occupied more than a half century, it may be of interest to note some of the now historic incidents attending this gradual change.

For many years antedating the Civil War there had been progressively some sort of recognition of the economic value of the expansive use of steam, until shortly before that war began it had grown, with some engineers of the day, into an unreasoning furore. They, as the admiral says, "hardly seemed to realize that there was any limit to expansion." In the words quoted, Admiral Melville probably had in mind, among the expansion extremists of the time, the late Frederick E. Sickels and his partner and *fidus Achates* E. N. Dickerson.

Dickerson was a patent attorney of superior note who had conducted for Sickels an infringement suit against the late Geo. H. Corliss, and through this finally blossomed out—in his own estimation at least—into a full blown marine engineer; and doubtless did much to encourage and intensify in Sickels his craze for high expansion.

Sickels was a most ingenious and prolific inventor, as the writer had ample opportunity to verify during the several months occupied by him in completing the "Pensacola's" engines, while at anchor in the Potomac; the government—suspecting Sickels' loyalty, although I am sure without the least reason therefor—having confined him to the ship until the machinery was finished, and the vessel had run the Confederate batteries on the river banks.

Thoroughly imbued as he was with the extreme expansion craze, he early saw the incompetence of the Stevens cut-off to arrest the steam admission until the piston had completed some-

thing more than half its stroke, invented his now celebrated adjustable drop cut-off, actuated from some part of the engine which moved coincidently with the piston, in which way he could, in furtherance of his extreme expansion views, cut off at any point, however short.

While Sickels is entitled to the greatest credit for this invention as a means to the end he sought, his failure to see the futility of cutting off much, if any, short of half stroke under the then obtainable conditions governing the matter, but energetically pursuing his expansion phantom, obtained from the government just previous to the outbreak of the war a contract to equip the then new and magnificent sloop of war "Pensacola" with steam machinery, after his own designs.

After admission to the service, the writer was ordered to the engineer force of this ship, then lying at the docks at Washington far from completed, and if time and space here permitted it would doubtless be interesting to recount some of the many curious experiences, for an engineer, on this ship in her subsequent completion and in her career in the passage of Forts Jackson and St. Phillip on the lower Mississippi and the capture of New Orleans under the great Farragut. She never got further than New Orleans, however, where the writer was detached from her. She was always known as the lame duck of the fleet, and as an engineering product was probably the most monumental failure of her time; and all because the designer of her machinery failed to comprehend the limitations which engineering knowledge then placed upon the expansive use of steam.

Cursorily, I will point out that her main machinery consisted of four horizontal cylinders of 52 inches diameter, and four feet stroke, fitted with Sickels' adjustable drop cut-off actuating poppet valves. There were enormous clearance spaces in the steam chests and passages under this arrangement; in itself most inimical to the expansion idea, and to make matters worse, Sickels had contrived condenser apparatus so illy considered that we could rarely command more than 15 to 20 inches of vacuum at the best; then, to crown all, he—expecting expansion to do most of the business—provided boilers by far too small. Yet with a maximum allowable working pressure of only 35 pounds and all these drawbacks Sickels added, he expected to cut off at one-tenth of the stroke in these four large cylinders.

It will be readily seen that on this ship while under the guns

of the forts below New Orleans with 10 to 15 inches of vacuum and the boilers foaming so that for many minutes at a time we had no idea where the water level in them was, the life of an engineer was anything but a happy one.

Contrast with this the results of Isherwood's careful experimental methods and the ships he produced under them, and we have much to be thankful for that this honored member of ours was at the head of the Steam Engineering bureau during those critical and trying times.

Sickels and Dickerson, not convinced by their failures in the "Pensacola," succeeded in obtaining still another contract to equip one of the "double-enders," the "Algonquin," with machinery, having for its object the carrying out of the same extreme expansion ideas. Isherwood, in order to show that his experimental conclusions were better than Sickels' theoretics, took a sister ship, the "Winooski," fitted with an inclined engine, and the then somewhat notorious "Algonquin-Winooski" dock trial was had.

Sickels and Dickerson had succeeded in designing boilers in which they could safely carry 60 pounds, beginning to realize what an important factor initial pressure was to them. Isherwood had extra braces put in the Martin boilers of the "Winooski," so that they could safely carry 40 pounds. Sickels essayed to cut off about $\frac{1}{10}$; Isherwood had to be content with an initial pressure of 40 pounds, and cut off by the Stevens device at about half stroke. The writer was one of the watch engineers on the "Winooski" in this trial, and remembers that upon summing up, the results were that the "Winooski" with 40 pounds expanding once was considerably more economical than the "Algonquin" under the advantage of 20 pounds greater initial pressure expanding eight or nine times.

In those days no one fully understood the real causes which so greatly limited expansion possibilities. The great factor of cylinder condensation was far from being understood or appreciated; and particularly did we fail to see what an important factor in the production of this condensation was the question of piston speed. In the fifties 300 feet per minute was a good speed; now we quadruple that in short-stroke engines. Even Mr. Isherwood, with all his careful experimentation, up to within a year of the close of the war, did not realize the importance of high piston speed; otherwise he probably would not have geared the crankshaft to the propeller shaft of the "100 inch" cylinders of the "Wampanoag" class of ships.

Of course there were other elements, such as steam jacketting, super-heating, etc., all looking to a reduction of that cylinder condensation which was gradually forcing upon us a conviction of its inimical character, and these were more or less in use in those days. The many changes that have come since Sickels' time have finally brought about the successful use of steam expansively beyond his wildest dreams.

The experiments during, and the experiences of, the war seemed to show that the country was as fortunate—with marine engineering knowledge in such a transition state—in having so careful and painstaking an engineer as Mr. Isherwood at the engineering helm during those troublous times, as it since has been in having the services of a Loring and a Melville at the head of the Engineering Bureau while developing what we now know as our new navy.

Mr. A. Bement.—It is stated in paragraph 29 that English Engineers have had some trouble with the Belleville boiler, and the inference is that it was due to the lack of familiarity with it, but this boiler has been fitted to a very large number of ships in the British Navy, and it is hardly reasonable to attribute the trouble to lack of experience, considering the opportunity for becoming familiar with the device. The real trouble is with the boiler itself, rather than with the people who attempt to operate it. If criticism is justified, it is due in connection with the adoption of this apparatus rather than its management.

*Admiral Melville.**—I have to thank the gentlemen of the Society who so kindly discussed my paper and who dealt with it so leniently, as there is, no doubt, much in it to criticise.

I am delighted to hear from my old companion-in-arms of the Civil War period, J. T. Hawkins, whose reminiscences bring back to my mind many vivid memories of the scenes and experiences of the Civil War. He left our service much to the loss of our Navy, and has made his mark in the field of mechanical invention and in the promotion of the mechanic art.

I am also pleased to see the criticism of Mr. A. Bement, of Chicago. I am fully in accord with him in regard to his criticism of the Belville boiler, and I might add many others in the freak class both at home and abroad, but I did not think it proper to criticise any make of boiler either pro or con, as I do not believe it would be fair to the trade.

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* Author's Closure, under the Rules.

No. 1077.*

NOTE ON HEADS OF MACHINE SCREWS.

BY H. G. REIST, SCHENECTADY, N. Y.

(Member of the Society.)

1. The size and shape of machine screw heads as made by different manufacturers vary very greatly. In the case of screws made by some manufacturers the heads used on different sizes have different proportions, thus presenting a different appearance. The company with which I am connected is a large consumer of machine screws, and this variation has from time to time caused great inconvenience. Frequently designs are made to suit the screw heads of one manufacturer, and if afterwards it becomes desirable to purchase screws manufactured elsewhere, there may be an interference with other parts of the machine, due to the variation in size of heads. When fillister head screws were used with the space for the heads counterbored, it was necessary to make the counter-bore large enough to take the maximum size of head, thus not making a good appearance when smaller heads were used. The same was true in the case of flat-headed machine screws. The subject was taken up with several manufacturers of machine screws and it was found that they had no defined standard, but were open to suggestions, and were willing to supply screw heads to any ordinary dimensions without much if any extra cost. The extreme variation in dimensions found is shown graphically in the accompanying drawings by full lines, from which it will be seen that there is great irregularity, and, as stated above, the same manufacturers will in some cases supply larger heads proportionally than others. (See Figs. 142, 143 and 144.)

2. In order to establish uniformity in our own works, we prepared the accompanying formula and table giving the shapes and dimensions of the heads of the size of screws ordinarily used. The

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

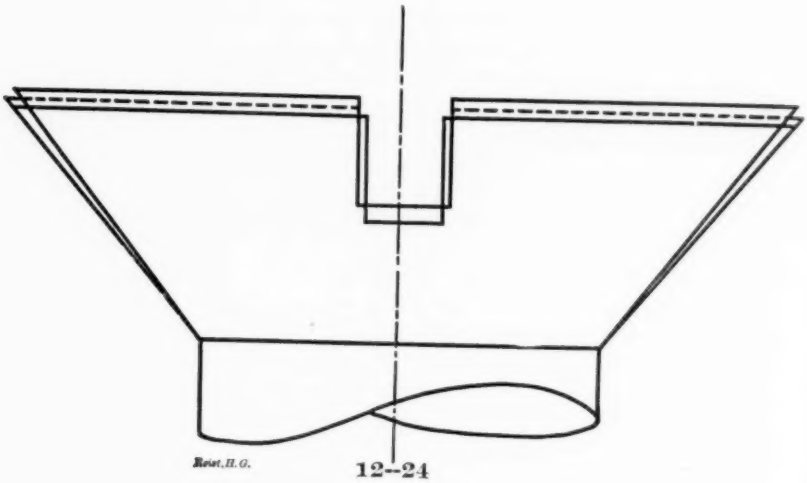


FIG. 142.

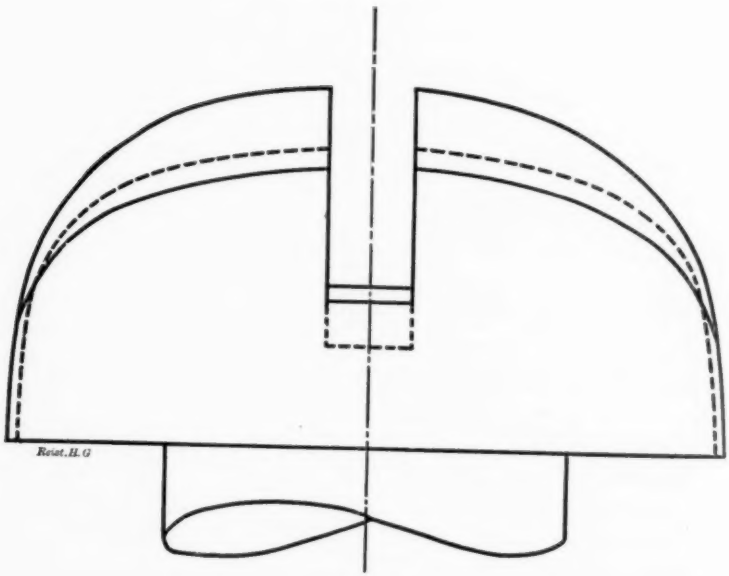
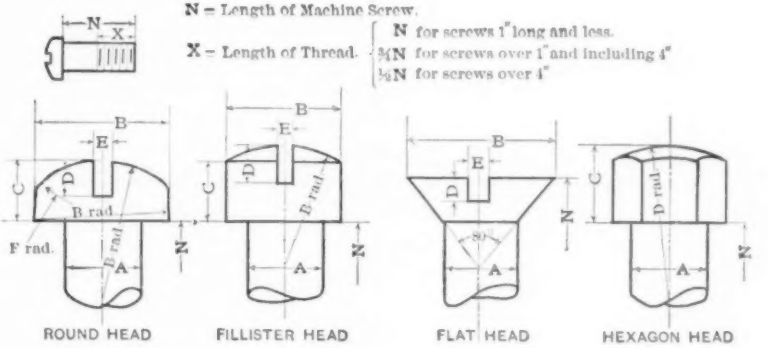


FIG. 143.

shape and proportional dimensions of these heads are shown by the broken lines in the drawings of the several forms of screw heads. It will be seen that these heads present a uniform appearance, and by submitting the table of dimensions we have had no difficulty in obtaining screws to the dimensions shown—in fact, the manufacturers have expressed themselves as pleased to have a standard to work to.



DIMENSIONS OF STANDARD MACHINE SCREWS.

Size of Screw.	No. of Threads.	Round Head Dimensions.						Fillister Head Dimensions.					Flat Head Dimensions.			
		A	*B*	*C*	*D*	*E*	*F*	*A*	*B*	*C*	*D*	*E*	*A*	*B*	*D*	*E*
0	.....	.058	.102	.044	.029	.015	.029	.058	.087	.044	.029	.015	.058	.116	.017	.015
1	.....	.071	.124	.053	.036	.018	.036	.071	.107	.053	.036	.018	.071	.142	.021	.018
2	64	.088	.154	.066	.044	.022	.044	.088	.132	.066	.044	.022	.088	.176	.026	.022
3	48	.099	.173	.074	.050	.025	.050	.099	.149	.074	.050	.025	.099	.198	.030	.025
4	36	.113	.198	.085	.057	.028	.057	.113	.170	.085	.057	.028	.113	.226	.034	.028
6	32	.141	.247	.106	.071	.035	.071	.141	.212	.106	.071	.035	.141	.282	.042	.035
8	32	.166	.291	.125	.083	.042	.083	.166	.249	.125	.083	.042	.166	.332	.050	.042
10	32	.190	.332	.143	.095	.048	.095	.190	.285	.143	.095	.048	.190	.380	.057	.048
12	24	.221	.387	.166	.111	.055	.111	.221	.332	.166	.111	.055	.221	.442	.066	.055
14	24	.246	.431	.185	.123	.062	.123	.246	.369	.185	.123	.062	.246	.492	.074	.062

		Hexagon Head Dimensions.			
		A	*C*	*D*	*G*
14	24	.246	.246	.492	.434

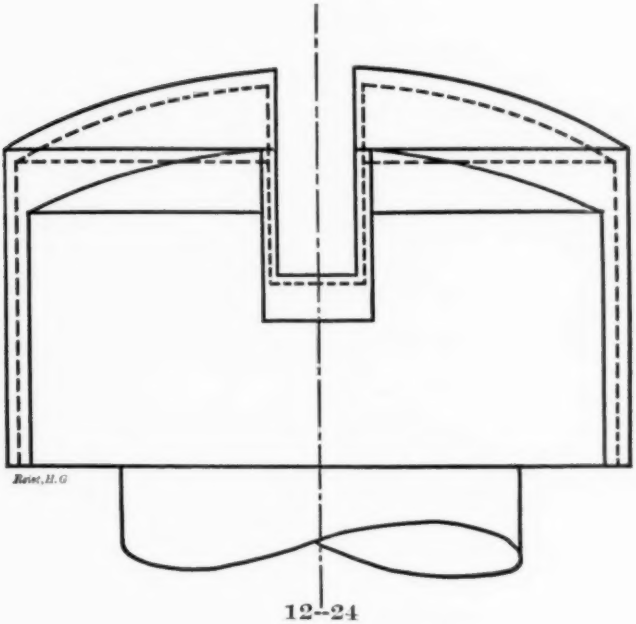


FIG. 144.

3. The author understands that there is a committee (appointed by the Society) at work on the standardization of machine screw threads, and it would seem to him that a report from them along these lines might make a valuable appendix to their work.

DISCUSSION.

Mr. L. D. Burlingame.—There is another style of screw head that has been extensively used which is not represented in the diagrams shown in this paper. It is similar to what is here called the fillister head screw, only it is made flat on top, with the corners rounded. It has some advantages over the styles here shown. In the works with which I am connected large quantities of screws with the head made in the style that I speak of have been used for many years. There are two advantages in using such a screw. One is that the counterbore does not have to be so uniform in depth in order to make a good appearance. Another is that this one style will take the place of the

first two styles illustrated on page 479, and can be used in places where either of these screws are used. Where a concern is carrying many thousand kinds and lengths of screws, any saving in the number of the kinds carried is certainly an item worth considering. Another point, that I would call attention to in connection with the formula given, is that making the diameter of the head one and a half times the body, while it may answer very well for the small screws which are listed in the table, seems inappropriate for large screws—say, up to 7/8 and 1 inch diameter. It makes the head too large in proportion to the body. We have found that a formula with a constant was much better; make the head one and one-quarter times the diameter of the screw, plus a sixteenth of an inch constant. Using this formula, the one-quarter inch screw, the largest in the list, would be practically the same as listed, but a three-quarter inch screw would have a head one inch in diameter instead of one and one-eighth inch diameter, thus giving a better proportion from the small to the large sizes. This formula applies well to hexagonal head screws. One single instance given in the paper of a hexagonal head with the diameter three-sixteenths of an inch larger than the body of the screw does not seem to be sufficient to establish a proportion suitable for a large number of sizes, while the formula I have mentioned could be used for large and small sizes.

The presentation of this paper is, it seems to me, made at an opportune time, and it could be considered, as suggested, by a committee to bring about a standard which could be generally used. It might not apply to such a factory as our own, where we make a large proportion of our own screws, but in supplying screws generally for the mechanical industries of the country it would certainly be a help to have some standard established. It would, however, seem wise to have it established along such lines as would apply to general conditions. The United States standard for the pitch of screw threads, which has been so extensively used, unfortunately has not seemed to apply to small machine screws, and most of us have had to depart from it in order to get the results we want. It would seem that some time a supplementary standard might be established which should include not only a standard for finer screw threads, but also proportions of heads and other parts of screws which could become the uniform American practice.

Mr. Whitehead.—I think it very important to have all these

screws standardized, and I agree with the gentleman who has just spoken and who has covered the ground very fully, that it is necessary to have all screw heads standardized in order that they may be manufactured more cheaply. The government standard screws have already been adopted as far as they can be for the work that they are adapted to. There are some classes of screws used which require finer threads than the government standard. My experience has not been very large on machine screws, as our business in that line has been a very small part of our plant, and has been confined more particularly to nuts and bolts. But the screw business is growing so fast I think it is a good thing to have all machine screws standardized.

I have not studied the subject carefully enough to be able to cover the ground any more fully than the last gentleman who spoke.

No. 1078.*

CAST IRON: CRUSHING LOADS AND MICROSTRUCTURE.†

BY W. J. KEEP, DETROIT, MICH.

(Member of the Society.)

1. The Crushing and Tensile Tests were made by H. Diederichs, Assistant Professor of Experimental Engineering, Sibley College, Cornell University. The Report on the Microstructure is by Albert Sauveur, Assistant Professor of Metallurgy, Harvard University, and proprietor of The Boston Testing Laboratories. The photomicrographs accompanying his report were taken and specimens were polished and etched in the Metallographical Laboratory of Harvard University by H. M. Boylston, S.B. The test specimens for all tests were prepared by the author.

2. The object of this paper is to make more complete the so-called A. S. M. E. series of tests on cast iron, which are recorded in the *Transactions* of this Society, in vols. xvi., xvii, xix., xxi., xxii. and xxv. (See second foot note.) It was intended at this time to present impact tests of these series of tests, but the special apparatus required and the peculiar results obtained made it advisable to delay its presentation.

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* Presented at the Scranton meeting, June, 1905, of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussions on this general subject, consult *Transactions* as follows:

No. 318, vol. 10, p. 187: "Tests of the Strength of Cast Iron." Gaetano Lanza.
No. 631, vol. 16, p. 542: "Relative Tests of Cast Iron." W. J. Keep.
No. 654, vol. 16, p. 1066: "Report of Committee on Standard Tests."
No. 655, vol. 16, p. 1082: "Transverse Strength of Cast Iron." W. J. Keep.
No. 695, vol. 17, p. 674: "Strength of Cast Iron." W. J. Keep.
No. 766, vol. 19, p. 351: "Cast Iron under Impact." W. J. Keep.
No. 842, vol. 21, p. 369: "Impact." W. J. Keep.
No. 892, vol. 22, p. 570: "Influence of Titanium on the Properties of Cast Iron and Steel." Auguste J. Rossi.
No. 1041, vol. 25, p. 884: "Cast Iron; Strength, Composition, Specification." W. J. Keep.

3. This series of tests is valuable on account of its completeness and because a full chemical analysis was made of each size of test-bar of each of the 19 series of casts.

4. One pair of test-bars of the Riehle shape 0.692 in. diameter, also one pair of the same shape 1.13 in. diameter, of each of the 19 series, were tested in tension by Prof. R. C. Carpenter of Sibley College, and recorded in vol. xvii. of the *Transactions*, pp. 726 and 727. A pair of bars, 1.13 in. diameter, from each series was ground true and tested in tension on an Emery testing machine by Prof. Ira H. Woolson, at Columbia University (ibid., pp. 728 and 729).

TABLE I.

CRUSHING TESTS OF SHORT CYLINDERS.

No. of Series.	Cylinders 0.692 in. Diam. × 2 in. Long.†					Cylinders 1.13 in. Diam. × 3.4 in. Long.§					Average per centage of Silicon.
	No. of Test Bar.	Crushing Load.		Length at Max. Load.	Compression per 1,000 lb.	No. of Test Bar.	Crushing Load.		Length at Max. Load.	Compression per 1,000 lb.	
		Maximum Load per sq. in.	Average.				Maximum Load per sq. in.	Average.			
1	....	.....	.....	.....	.....	22	.....	76,280	*	.....	0.80
2	....	.....	.....	.....	.....	43	76,200	.....	3.18	.....	.....
						44	72,000	74,100	3.20	.002834	1.21
3	11	84,906	.....	1.86	.....	65	76,440	.....	3.20	.....	.....
	12	84,053	84,480	1.87	.004258	66	78,000	77,220	3.24	.002344	1.88
4	....	.....	.....	.....	.....	87	69,440	.....	3.16	.....	.....
	16	.....	80,203	1.90	.003391	88	65,240	67,340	3.20	.002967	2.01
5	19	71,040	.....	1.89	.....	109	62,320	.....	*	.....	.....
	20	65,786	68,413	1.87	.005652	110	58,200	60,260	3.23	.002821	3.19
6	23	79,653	.....	1.87*	.....	131	65,200	.....	3.14	.....	.....
	24	81,648	80,651	1.86	.004463	132	58,360	61,780	3.14	.004208	3.04
7	27	77,760	.....	1.86*	.....	153	73,480	.....	3.19	.....	.....
	28	79,253	78,506	1.85*	.004926	154	67,160	70,320	3.20	.002915	0.93
8	31	76,133	.....	1.88*	.....	.....	.....	.....	.....	.....	.....
	32	75,573	75,853	1.90	.003883	175	.....	66,700	3.21	.002849	1.17
9	....	.....	.....	.....	.....	197	71,300	.....	*	.....	.....
						198	69,640	70,470	3.21	.002696	1.67
10	....	.....	.....	.....	.....	219	59,720	.....	*	.....	.....
						220	69,400	64,560	3.22	.002788	2.23
11	....	.....	.....	.....	.....	241	.....	68,680	3.19	.003575	2.71
12	47	61,680	.....	1.86	.....	263	57,200	.....	3.17	.....	.....
	48	63,733	62,706	1.80	.007229	264	59,000	58,100	3.17	.003959	3.50
14	55	77,253	*	1.90	.....	295	68,300	.....	3.20	.....	.....
	56	75,786	76,520	1.89	.003639	296	69,780	69,040	3.18	.003541	2.82
13	51	74,133	.....	1.90	.....	275	77,840	.....	*	.....	.....
	52	80,720	77,426	1.89	.003617	276	68,500	73,170	3.22	.002460	3.18
15	59	71,680	.....	1.86	.....	307	76,000	.....	3.10	.....	.....
	60	71,520	71,600	1.85	.005400	308	72,940	74,470	3.15	.003693	3.50
16	63	85,040	.....	1.86	.....	.....	.....	.....	.....	.....	.....
	64	84,880	84,920	1.88	.004082	.....	.....	.....	.....	.....	1.76
18	67	107,413	.....	1.87	.....	65	90,920‡	.....	3.18	.....	.....
	68	105,200	106,306	1.88	.003136	66	80,000	85,460	3.18	.002584	2.06
19	....	.....	.....	.....	.....	73	88,680‡	.....	3.21*	.....	.....
						74	82,800	85,740	3.20*	.002273	0.77

† All cylinders 0.692 in. diameter, were cut from the tensile bars of Table xxvii, p. 726, vol. xvii. *Transactions* A. S. M. E. ‡ Cylinders 65 to 74, 1.13 in. diameter, were from tensile bars of Table xxviii, p. 727. *ibid.* For tensile tests of Series 18, see test bars 73 to 83, and for Series 19, see test bars 103 and 104, Table xxxii, p. 729, *ibid.* § All cylinders 1.13 in. diameter, except for Series 18 and 19, were cut from transverse tests bars cast 1.13 in. diameter, Table xxiv, p. 723, *ibid.* * Short column failure, all others long column failure. For a full chemical analysis of all these series, see pp. 1100 to 1107, vol. xvi, *ibid.*

5. All of the original test-bars are carefully preserved for future tests, consisting of four bars 13 in. long of each size from 1 in. to 4 in. square of each of the 19 series.

6. Table I. contains crushing tests of bars, 0.692 in. and 1.13 in. diameter, to correspond with the tensile tests already referred to of the same sectional area and of the same iron.

7. The length of all of these test-pieces for crushing was three times the diameter. They were carefully turned to size with end faces parallel and bedded with sheet copper.

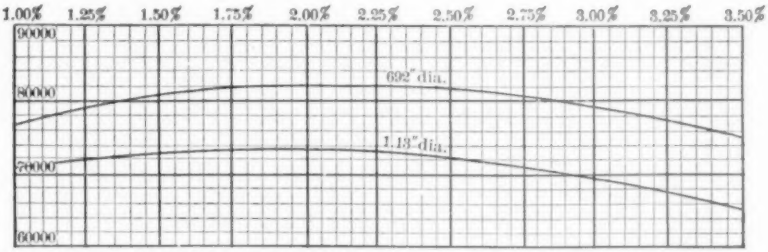


FIG. 145.—CHART OF CRUSHING STRENGTH SHOWING APPROXIMATE RELATION OF STRENGTH TO SIZE OF CASTING AND TO THE PERCENTAGE OF SILICON
(Chart gives the strength per square inch.)

8. From these records the chart (Fig. 145) was prepared, showing the same general variation in crushing strength for variations, in size and in silicon, that was found in transverse and tensile strength. (See vol. xxv., pp. 895 and 898.)

Physical Composition of Castings.

9. The variation in structure between the edge and the center of a round casting, and the corner, side and center of a rectangular casting, is shown in Tables II., III. and IV. While there does not seem to be any difference in the size of the grain under the microscope, even when magnified 500 diameters, yet there is a very great difference in strength.

10. All test-pieces in Tables II., III., IV. and VI. were cut from bars of one square inch area, of Series 2, with silicon 1.21 per cent., and all were poured from iron taken from the cupola at the same time.

11. For another interesting test record, a block of cast iron 9 inches square by 18 inches long was cut longitudinally into 9 pieces, and test-bars, 1.13 in. diameter, were turned from each piece. The average tensile strength from the bars from the corners

was 25,100 lb., from the sides 19,913 lb. and from the center 15,730 lb. (Vol. xxv., p. 914.)

TABLE II.

Crushing Tests of ½-in. Cubes.	Maximum Load.		
	Actual.	Lb. per sq. in.	Av. per sq. in.
½-in. cubes from corner of 1-in. □ test bar.....	21,760 21,970	87,040 87,880	 87,460
½-in. cubes from side of 1-in. □ test bar	18,780 18,990	75,120 75,960	 75,540
½-in. cubes from center of 1-in. □ test bar.....	17,120 17,410	68,480 69,640	 69,060
½ in. deep × 1 in. □.....	.....	.....	112,700

TABLE III.

Tensile Tests of ½-in. □* Bars.	Maximum Load.		
	Actual.	Lb. per sq. in.	Av. per sq. in.
½-in. □ × 6-in bars from corner of 2 in. × 1-in. bars.....	3,700 3,280	20,453 18,131	 19,192
½-in. □ × 6-in. bars from side of 2-in. × 1-in. bars.....	3,520 3,410	19,453 18,297	 18,870
½-in. □ × 6-in. bars from center of 2-in. × 1-in. bars	3,520 3,270	19,453 18,076	 18,764

* Three inches at the center was turned down to a diameter of 0.480 in. We used 2-in. x 1-in. bars because we were short of 1-in. □ bars of Series 2.

TABLE IV.

Crushing Strength per sq. in. of ½-in. Cubes cut from each Size of □ Dry Sand Bars of Series C of Tests by the American Foundrymen's Association.

	Distance of Cube from Surface.	½ in. □	1 in. □	1½ in. □	2 in. □	2½ in. □	3 in. □	3½ in. □	4 in. □
A. F. A. Series C, Si. 2.01 %.	First ½ in.....	154,000	111,600	88,240	87,000	79,200	72,080	68,400	65,640
	Second ".....	.....	.....	83,000	77,360	72,200	67,400	66,040	61,000
	Third ".....	.....	.....	.....	.....	71,360	64,160	64,320	59,520
	Fourth ".....	.....	.....	.....	.....	.....	.....	63,520	56,800
	Center ".....	154,000	99,560	83,000	72,040	71,360	63,800	63,520	56,880
	Per sq. in. of bar.....	154,000	*107,588	86,492	80,152	74,832	67,468	65,864	60,312

* The crushing load per sq. in. is found by taking twice the sum of the loads on cubes outside the center + the load at the center, and recording the average.

TABLE V.

Crushing Tests—Sections of 1-in. □ Test Bars.	Maximum Load.		Length at Max. Load.	Compression per 1,000 lb.
	Actual.	Average per sq. in.		
Cylinder 1.13 in. diameter × 3.4 in. long	76,200	.....	3.18	.....
	72,000	74,100	3.20	.002834
1 in. □ × 3 in. long	†.....	(79,700)	(2.83)	(.002133)
	76,000	76,000	2.91	.001071
Cylinder 1 in. diameter × 3 in. long	60,480	77,020	2.80	.003307
Cylinder ½ in. diameter × 2½ in. long	30,800	69,710	2.18	.002272

† See note Table VI.

TABLE VI.

Crushing of Columns.	Maximum Load per sq. in.	Length at Max. Load.	Compression per 1,000 lb.	Compression per inch per 1,000 lb.
1 in. □ × ½ in. long	*112,700	0.38	.001064	.002128
1 in. □ × 1 in. long	*96,420	0.87	.001347	.001347
1 in. □ × 1½ in. long	*93,030	1.37	.001397	.000931
1 in. □ × 2 in. long	*80,600	1.85	.001861	.000930
1 in. □ × 3 in. long	† (79,700)	(2.83)	(.002133)	.000711
	* 76,000	2.90	.001316	.000438
1 in. □ × 4 in. long	78,850	3.77	.002662	.000665
1 in. □ × 8 in. long	‡63,380	7.72	.004419	.000552
1in.□ × 12 in. long	§47,570	11.80	.004204	.000350

* Short column fracture; all others long column fracture. † This record should have been about 79,700; the piece was defective. ‡ Began bending at 48,000 lb. § Began bending at 43,000 lb.

12. Table V. gives the strength of a test-piece cast and tested 1.13 in. diameter, one cast and tested 1 in. □, and another cast 1 in. □, but which had the corners turned off, and still another with

corners and sides removed. The difference in strength between a small and a large casting is shown in Table I. and in Figs. 178 to 187.

13. The reason for dwelling upon this at such length is that very few realize that a variation in the size of a casting causes a

TABLE VII.

Crushing Steel, Test Pieces 3 in. Long.	At Yield Point.			Crushing Load.		
	Load per sq. in.	Length at Load.	Compression per 1,000 lb.	Load per sq. in.	Length at Maxim'm Load.	Compression per 1,000 lb.
1-in. ○ soft steel...	34,283 35,495	2.95 2.90	.001786 .003449	69,282 69,008	2.57 2.58	.007602 .007432
1-in. □ Swede iron.	36,000 36,000	2.98 2.97	.000555 .000833	100,000 75,200	1.85 2.40	.010500 .007979
1-in. □ soft steel ..	38,000 36,000 36,000	2.95 2.99 2.98	.001316 .000279 .000555	80,460 84,900 75,600	2.60 2.52 2.69	.004971 .005654 .004101

TABLE VIII.

Crushing Columns, Steel Bars 10 in. Long, Compression in 8 Inches. Held in Grips.	At Yield Point.			Maximum Load.		
	Load per sq. in.	Length at Load.	Compression per 1,000 lb.	Maximum Load per sq. in.	Length at Maxim'm Load.	Compression per 1,000 lb.
1-in. ○ soft steel ..	33,111 33,111	7.99 7.99	.000375 .000375	35,778 38,197		
1-in. □ Swede iron.	22,000	7.99	.000284	25,500		
1-in. □ soft steel ..	34,000 34,000	7.99 7.99	.000348 .000348	44,200 * 41,000 *		

* At 36,000 buckling and scaling were very plain.

greater variation in strength than a variation in chemical composition; and that whatever size of test-bar is used, the strength of any other size of casting must be approximated by a comparison with a series of tests of such sizes of castings, and cannot be calculated by a mathematical formula. (See charts and tables for approximating transverse and tensile strength, vol. xxv., pp. 895, 898 and 901.)

Best Form of Test-piece for Crushing Tests.

14. The specimen to be tested should be cast one square inch in area and 1.13 in. diameter. It will not do to cut a test-piece from a larger casting, because wherever cut it will not represent the structure of such casting.

15. A length proportioned to the diameter should be fixed upon as standard. The conditions in Tables VII. and X. seem best.

TABLE IX.

Columns in Tension. Steel Bars 10 in. Long, Extensions in 8 Inches. Held in Grips.	At Yield Point.			Maximum Load.		
	Load per sq. in.	Length at Load.	Extension per 1,000 lb.	Maximum Load per sq. in.	Length at Maxim'm Load.	Extension per 1,000 lb.
1-in. ○ soft steel...	34,380	8.06	.002222	50,840	9.81	.035602
1-in. □ Swede iron.	20,000	8.02	.001020	41,900	8.46	.04177
1-in. □ soft steel..	32,000	8.01	.002453	58,370	9.90	.03235

TABLE X.

No. of Series.	1 " area, 8 in. — Crushing Tests.	% C°.C.	% Si.	Compression at Load of			Maximum Crushing Load.
				14,000	16,000	22,000	
	□ Soft steel		0.	.00295	.00345	.00490	44,200
17	○ White cast iron	2.85	0.81	.00400	.00465	.00645	123,520
19	○ Carwheel iron.....	0.99	0.72	.00570	.00650	.00955	61,211
18	○ Heavy machinery	0.52	2.09	.00330	.....	.....	.....
13	○ Stove plate iron	0.10	3.22	.00730	.00855	.01325	48,960

Relation of Strength to Length of Test-piece.

As there were not enough round bars in Series 2 for all crushing tests 1 in. □ bars were used for Table VI.

16. Referring to Table II., we see that the strength of a ½-in. cube, cut from the center of a test-piece 1 in. □, is 69,060 lb. per sq. in.; when cut from the corner the strength is 87,460 lb. per sq. in., but when a section 1 in. □ × ½-in. thick is cut clear across the test-bar and is tested, it gives 112,700 lb. per sq. in. A ½-in.

cube from the side of a test-bar 4 in. □ gives 56,880, but a section 4 in. □ × ½ in. thick would show a greater strength per sq. in. than the section 1 in. □ × ½ in. thick from a test-bar 1 in. □.

The outside metal in a larger section prevents the metal at the center from being crushed.

17. Table VI. shows that as the length increases the strength decreases. All specimens less than 3 inches long fail as a short column—that is, do not bend but break in pieces. From 2½ to 3 times the diameter has generally been considered best.

TABLE XI.

No. of Series.	Tensile Tests. (1 □" area, 8 in.)	% C[d].C.	% Si.	Extensions at Load of 11,000	16,000	22,000	Maximum Tensile Load.
	□ Soft steel		0.	.00305	.00365	.00525	58,370
17	○ White cast iron	2.85	0.81	.00400	.00490	.00745	24,000
19	○ Carwheel iron	0.99	0.72	.00310			30,000
18	○ Heavy machinery	0.52	2.09	.00500			24,000
13	○ Stove plate iron	0.10	3.22	.01460	.01995		16,850

18. The bending that takes place in long test-bars when the crushing force is applied to flat ends may be largely avoided if the test-bar for the crushing test is held in the grips of the machine the same as the tensile bar is held. (See Tables VIII. to XI.)

19. A comparison of Tables VII. and VIII. will show the results of crushing steel bars 3 inches long and 8 inches long.

All test-pieces in Tables VII. to XI. having the same name are cut from the same rolled bar of soft steel or Swede iron.

20. Table IX. shows the tensile strength of the same test-bars.

21. Tables X. and XI. show that testing-bars with 8 inches on which to measure extensions or compression and held in the same grips shows the tensile and crushing strength much better than by using shorter test-bars.

Microstructure of Cast Iron.

22. In each cast, or series, a number of test-bars of one square inch section, some square and others round, were made in molds and of iron of as uniform quality as would be possible in any foundry.

23. Some test-bars of 1.13 inch diameter were tested trans-

versely and in tension; all 1 in. □ bars were tested transversely and a complete chemical analysis was made of all sizes of square test-bars.

24. The photo-micrographs (Figs. 149 to 153) of the center and edge were made from a specimen cut from test bar No. 43, 1.13 in. diameter, and Figs. 154 to 159 of the corner side and center were made from a specimen cut from one of the 1 in. □ test-bars, all from Series 2.

25. Series 15 (and 13) was a regular high silicon stove-plate mixture of 80 tons per day.

26. Series 18 was in 1894 considered the best mixture for heavy machinery of Messrs. Bement, Miles & Co., Philadelphia, Pa. For a description see *Transactions*, vol. xvi., p. 1078. Figs. 178 and 179 are from the center of the test-piece 0.692 in. diameter. Figs 180 to 187 are $\frac{1}{2}$-in. from the edge of the test-bar.

27. Series 19 was in 1894 a good car-wheel mixture of Messrs. A. Whitney & Sons of Philadelphia (Figs. 188 and 189).

Mild steel (Fig. 191) is from the square bars of Tables VII. to XI.

Report of Professor Albert Sauveur.

28. Photomicrographs were made by Mr. Boylston from the unetched surfaces because such surface shows the graphite particles better than an etched surface, and the mode of occurrence of graphite particles have a marked influence upon the properties of the iron.

29. In order to justify inferences arrived at in every case it will be well to bear in mind the two following propositions:

First. *For a given composition* of cast iron the smaller the particles of graphite the stronger the iron.

Second. *A cast iron of maximum strength* should have a metallic matrix of maximum strength, and this means that this matrix should consist only of pearlite or should contain but a slight excess of cementite, which in turn means that the proportion of combined carbon should be somewhere between 0.7 per cent. to 1.00 per cent.

30. There is no marked difference in structure between the center, the edge and the corner, nor is this to be expected because the rate of cooling of so small a bulk of iron, cast in sand moulds, should not vary much.

31. In the specimen from bar 59, 0.692 in. diam. of series 15

SERIES 1.—CENTER 1.13 IN. ○.

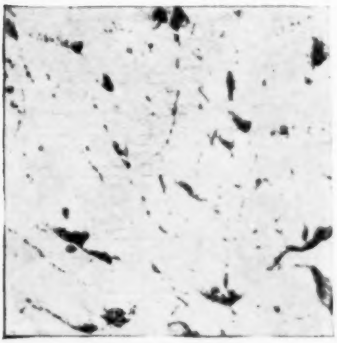


FIG. 146.—UNETCHED 100 DIAMETERS.

SERIES 2.—CENTER 1.13 IN. ○.

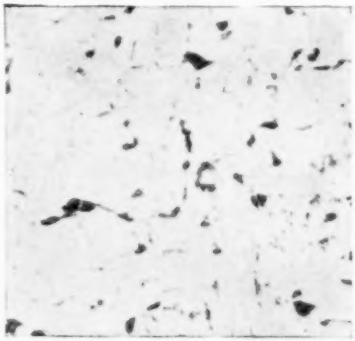


FIG. 149.—UNETCHED 100 DIAMETERS.

FIG. 147.—ETCHED 100 DIAMETERS.

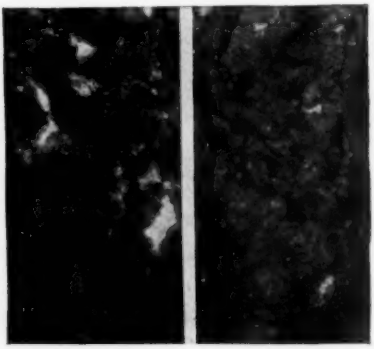


FIG. 150.—CENTER. FIG. 151.—EDGE.
Etched 100 Diameters.

For 1,000 Diameters, see Fig. 190.
FIG. 148.—ETCHED 500 DIAMETERS.

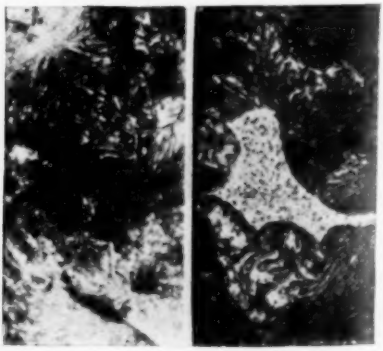


FIG. 152.—CENTER. FIG. 153.—EDGE.
Etched 500 Diameters.

CENTER. SERIES 2.—1 IN. □. CORNER.

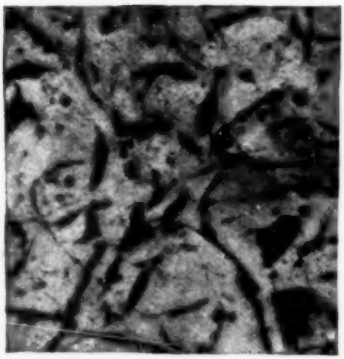


FIG. 154.—UNETCHED 100 DIAMETERS.

FIG. 157.—UNETCHED 100 DIAMETERS.

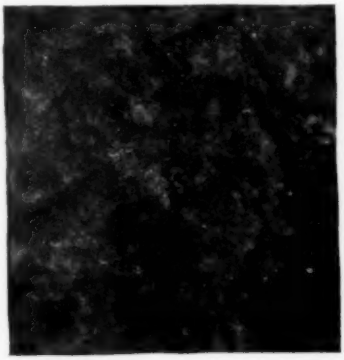


FIG. 155.—ETCHED 100 DIAMETERS.

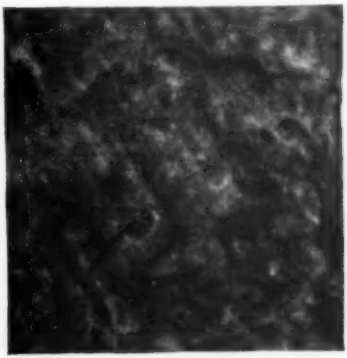


FIG. 158.—ETCHED 100 DIAMETERS.

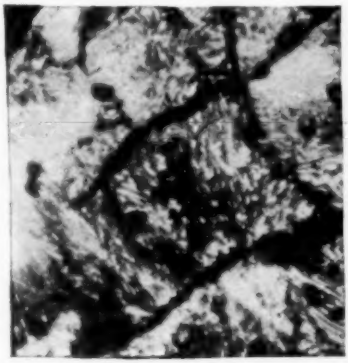


FIG. 156.—ETCHED 500 DIAMETERS.

Side.

FIG. 159.—ETCHED 100 DIAMETERS.

SERIES 3.—CENTER 1.13 IN. ○.

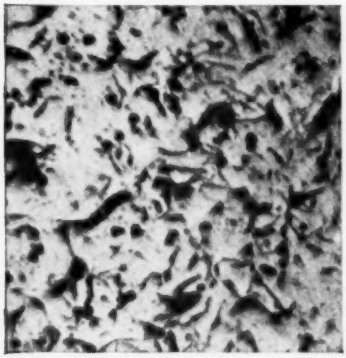


FIG. 160.—UNETCHED 100 DIAMETERS.

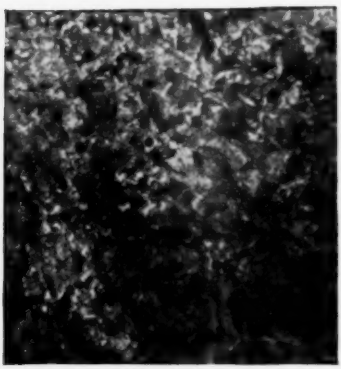


FIG. 161.—ETCHED 100 DIAMETERS.

FIG. 162.—ETCHED 500 DIAMETERS.

SERIES 4.—CENTER 1.13 IN. ○.

FIG. 163.—UNETCHED 100 DIAMETERS.

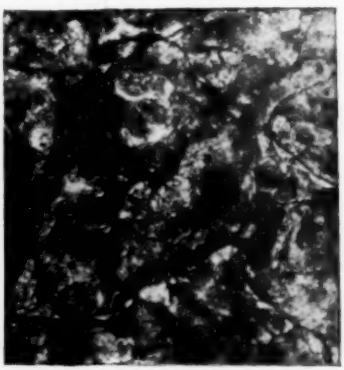


FIG. 164.—ETCHED 100 DIAMETERS.

FIG. 165.—ETCHED 500 DIAMETERS.

SERIES 5.—CENTER 1.13-IN. ○.

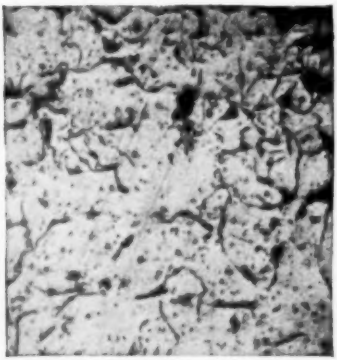


FIG. 166.—UNETCHED 100 DIAMETERS.

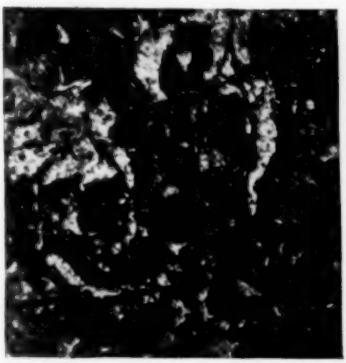


FIG. 167.—ETCHED 100 DIAMETERS.

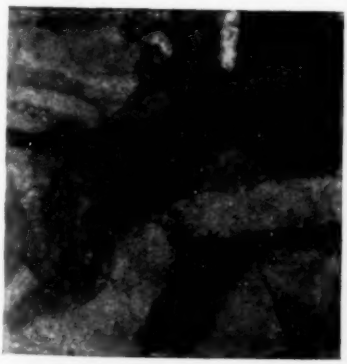


FIG. 168.—ETCHED 500 DIAMETERS.

SERIES 6.—CENTER 1.13-IN. ○.

FIG. 169.—UNETCHED 100 DIAMETERS.

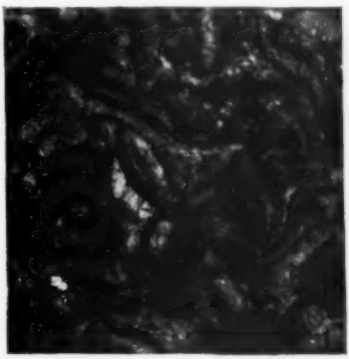


FIG. 170.—ETCHED 100 DIAMETERS.

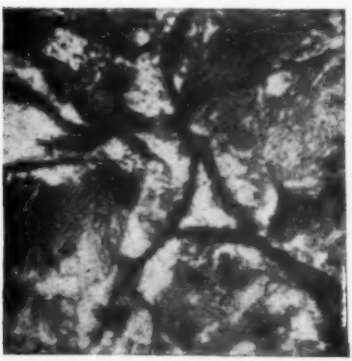


FIG. 171.—ETCHED 500 DIAMETERS.

0.692 IN. ○. SERIES 15.—CENTER. 1.13-IN ○.

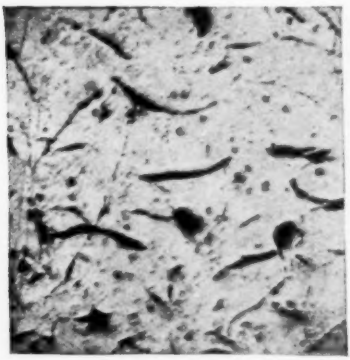


FIG. 172.—UNETCHED 100 DIAMETERS.

FIG. 175.—UNETCHED 100 DIAMETERS.

FIG. 173.—ETCHED 100 DIAMETERS.

FIG. 176.—ETCHED 100 DIAMETERS.

FIG. 174.—ETCHED 500 DIAMETERS.

FIG. 177.—ETCHED 500 DIAMETERS.

0.692-IN. ○. SERIES 18.—CENTER. 1.13-IN. ○.

FIG. 178.—UNETCHED 100 DIAMETERS.

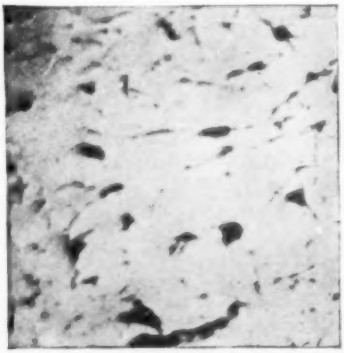


FIG. 180.—UNETCHED 100 DIAMETERS.

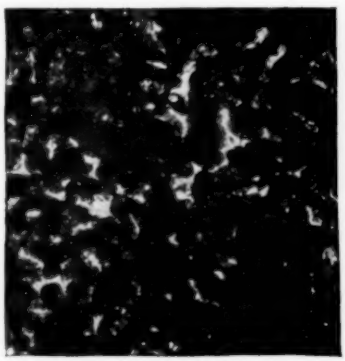


FIG. 179.—ETCHED 100 DIAMETERS.

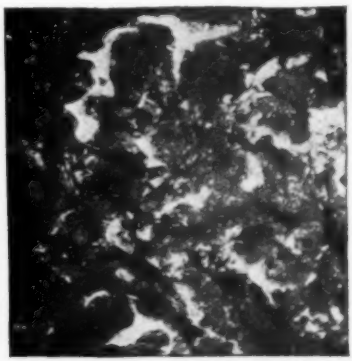


FIG. 181.—ETCHED 100 DIAMETERS.

2 in. □.

FIG. 182.—UNETCHED 100 DIAMETERS.

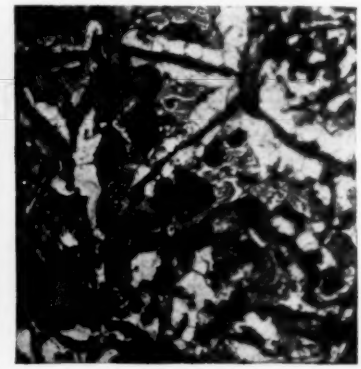
2 in. □.

FIG. 183.—ETCHED 100 DIAMETERS.

SERIES 18.—CENTER.

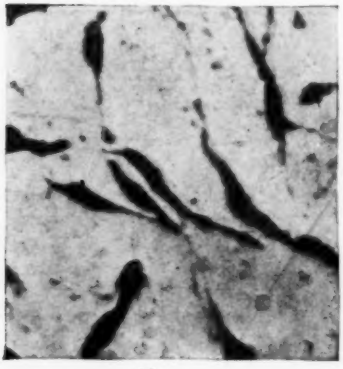
3-in. □.

FIG. 184.—UNETCHED 100 DIAMETERS.



3-in. □.

FIG. 185.—ETCHED 100 DIAMETERS.



4-in. □.

FIG. 186.—UNETCHED 100 DIAMETERS.

4-in. □.

FIG. 187.—ETCHED 100 DIAMETERS.

SERIES 19.—CENTER 1.13-IN. ○.

FIG. 188.—UNETCHED 100 DIAMETERS.

SERIES 19.—CENTER 1.13-IN. ○.

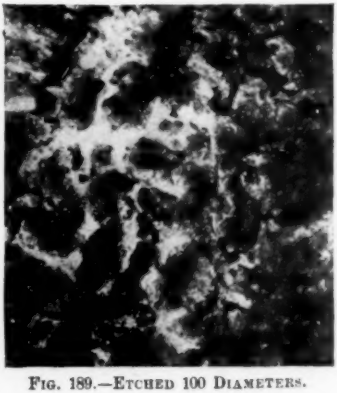


FIG. 189.—ETCHED 100 DIAMETERS.

SERIES 1.—CENTER 1.13-IN. ○.

FIG. 190.—ETCHED 1,000 DIAMETERS.
For 100 and 500 diameters, see Figs. 146, 147 and 148.

SOFT STEEL, 1-IN. □.

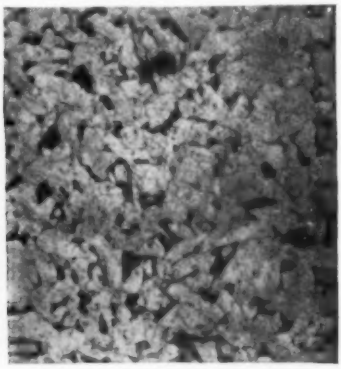


FIG. 191.—ETCHED 100 DIAMETERS.

the difference between the combined carbon reported by the chemist and as indicated by the structure is extreme. The chemist reports for a test-bar, 0.5 in. □, 0.10 per cent. of combined carbon while the structure shows about 0.4 per cent.

32. It is very surprising that notwithstanding this low amount of combined carbon and the fact that the graphite particles are quite coarse, the tensile strength of bar 59 is 24,400 lb. per sq. in. As shown from the structure, we should have inferred that it would have been considerably below 18,000 lb. per sq. in.

33. It should be borne in mind that the specimens of cast iron which were subjected to a microscopical examination were neither those that were analyzed nor those that were subjected to the physical tests. It is not to be expected, therefore, that the photomicrographs which we here reproduce correspond to the structure of the bars actually tested, neither do they suggest accurately the chemical composition of these bars. To try to infer the physical or chemical characteristics of samples of cast iron from the structure of *other samples*, although cast at the same time and under the same conditions, would be a very hazardous proposition, and it is why it was not attempted in this report.

34. The disagreement which exists between the structures here shown and the results of the physical test and the chemical composition of other bars is not to be wondered at considering the difference which generally exists between test-bars cast under identical conditions and from the same heat.

TABLE XII.

Prof. Sauveur's Microanalysis.

No. of Series.	No. of Test Bar.	No. of Figure Photomicrograph.	Size of Test Bar.	Condition of Graphite Particles.	Apparent % of C.	Tensile Strength per sq. in. From Structure.	Tensile Strength per sq. in. Prof. R. C. Carpenter's Test.
1	22	2, 3, 4, 46	1.13 in. ○	Small and fine.	.80	Exceed 20,000	15,700
2	43	5, 6, 7, 8, 9	1.13 in. ○	Finely divided.	.80	" 20,000	22,500
	.....	10, 11, 12, 13, 14, 15	1 in. □	Quite coarse.*	.80	" 20,000 (but less than 22 & 43)	
3	65	16, 17, 18	1.13 in. ○	Relatively coarse.	.60	Less than 20,000	20,450
4	87	19, 20, 21	1.13 in. ○	Smaller than 65.	.60	" " 20,000	19,350
5	109	22, 23, 24	1.13 in. ○	Small.	.40	" " 19,000	19,750
6	131	25, 26, 27	1.13 in. ○	Very coarse.	.45	" " 18,000	17,200
15	59	28, 29, 30	.692 in. ○	Quite coarse.	.40	" " 18,000	†24,400
	307	31, 32, 33	1.13 in. ○	Medium.	.10	" " 17,000	20,300
18	67	34, 35	.692 in. ○	Small and fine.	.10	Exceed 20,000	
	65	36, 37	1.13 in. ○	" " "	.70	" 20,000	24,110
	338	38, 39	2 in. □	Very coarse.	.20	Less than 15,000	
	340	40, 41	3 in. □	Coarse.	.25	" " 16,000	
	341	42, 43	4 in. □	Very coarse.	.15	" " 14,000	
19	73	44, 45	1.13 in. ○	Coarse.	1.10	Exceed 22,000	31,545

Mild steel, 47.

* No marked difference in the appearance of the grain at the edge and center of a round bar or the corner side and center of a square bar.

† The photomicrograph was from the sample that was tested in tension, therefore in this case a comparison can be made.

TABLE XIII.

Chemical Composition for Comparison.

No. Series.	Size Bar.	T. C.	G. C.	Cᵇ. C.	Si.	S.	P.	Mn.	Transverse Strength per sq. in.
Iroquois		4.05	3.20	0.87	.98	.035	.225	.49	Mixed to make Series 1 to 6.
Pencost...		2.79	2.04	0.75	10.00	.015	.487	.67	
1	1 in. □	3.85	2.60	1.25	.79	.054	.213	.36	2,025
2	1 in. □	3.86	3.32	.54	1.14	.040	.269	.30	2,046
3	1 in. □	3.75	3.28	.45	1.73	.030	.267	.51	2,540
4	1 in. □	3.72	3.24	.48	1.69	.046	.283	.30	2,410
5	1 in. □	3.56	3.40	.16	2.65	.021	.330	.39	2,464
6	1 in. □	3.46	3.08	.38	2.69	.033	.299	.45	2,214
15	.5 in. □	3.13	3.03	.10	3.29	.088	.980	.50	3,392
	1 in. □	3.10	3.01	.09	3.50	.090	.975	.49	2,235
18	.5 in. □	3.35	2.90	.45	2.39	.052	.340	.35	3,568
	1 in. □	3.35	2.83	.52	2.09	Drillings of all sizes of test bars were mixed and analyzed in Series 18 and 19.			2,250
	2 in. □	3.30	3.06	.24	1.82				2,000
	3 in. □	3.23	3.11	.12	2.06				1,877
	4 in. □	3.31	3.20	.11	1.88				1,835
19	1 in. □	3.85	1.86	.99	.72	.101	.353	.35	2,880

Chemical determinations by Messrs. Dickman & Mackenzie, Chicago, Ill. It was intended that Series 1 to 6 should contain 1.00, 1.50, 2.00, 2.50, 3.00, 3.50 per cent. silicon, but no excess was added for loss in melting. Other discrepancies are due probably to uneven diffusion.

NOTE BY W. J. K.—In Table xii the tensile tests by Professor Carpenter and the two footnotes, and Table xiii are not a part of Professor's Sauveur's report.

It was hoped that the photomicrographs would explain variations in the records of physical properties and further examination may do this to some extent.

It was thought that a comparison of the crushing tests of this paper with the tensile tests previously made would show the relation between extension and compression for varying percentages of silicon, but the crushing of short cylinders fails to show this; and the very satisfactory tests shown in Tables viii to xi were made on long test bars held in grips with 8 inches on which to measure changes in length.

It is expected that an apparatus will be perfected that will simultaneously make autographic diagrams of transverse load and deflection, and of elongation and compression of the extreme fibres; and that from these data can be calculated by a formula the position of the neutral axis and the crushing and tensile strength. Probably these results can be presented to the Society in the near future.

No. 1079.*

STANDARD UNIT OF REFRIGERATION.†

BY F. E. MATTHEWS, NEW YORK, N. Y.
(Junior Member of the Society.)

1. In presenting the following paper suggesting an outline by which to establish a standard unit of refrigeration the writer wishes to call attention more particularly to the principles involved and the methods employed than to any given example or specific computation.

The following out of lines parallel to those by which boiler ratings were established, commends itself both in that it lends consistency to the work of the society and makes the subject more tangible to the engineer less familiar with the details of refrigeration than steam engineering.

The numerical data of the paper conform to some extent to the recommendations already made by the committee appointed to

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* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussion on this general subject, consult *Transactions* as follows:

No. 353, vol. 10, p. 792: "Performance of a Thirty-five Ton Refrigerating Machine of the Ammonia Absorption Type." J. E. Denton.

No. 389, vol. 11, p. 830: "Test of a Refrigerating Plant." De Volson Wood.

No. 492, vol. 13, p. 507: "Summary of Results of Principal Experimental Measurements of Performance of Refrigerating Machines." J. E. Denton and D. S. Jacobus.

No. 514, vol. 14, p. 183: "Notes on the Refrigerating Process and its Proper Place in Thermodynamics." Geo. Richmond.

No. 554, vol. 14, p. 1414: "The Refrigerating Machine of To-day." C. Linde.

No. 707, vol. 18, p. 127: "Method of Determining the Work Done by a Refrigerating Plant and its Cost." F. H. Boyer.

No. 1025, vol. 25, p. 292: "Standard Unit of Refrigeration." J. C. Bertsch.

No. 1055, vol. 26, p. 64: "Preliminary Report of the Committee Appointed to Suggest a Standard Tonnage Basis for Refrigeration."

suggest a standard tonnage basis for refrigeration and in that respect is not wholly original.

1. *Standard Thermal Unit.*

Ton of refrigeration, equal to 288,000 British thermal units of negative heat, equivalent to the melting of 2,000 pounds of ice from and at 32 degrees Fahr., the latent heat of ice being taken as 144 British thermal units.

2. *Equivalent Standard Unit.*

Under *standard conditions*, the evaporation to a saturated vapor, of a sufficient number of pounds of a refrigerating fluid *per hour* to absorb heat at the rate of 288,000 British thermal units per twenty-four hours.

(1) *Standard Conditions.*

Since the evaporation of a liquid "from and at" its boiling point involves only the latent heat of volatilization, the adoption of a "from and at" basis similar to that used in the case of steam seems advisable. More specifically the number of pounds of the respective refrigerating fluids to be evaporated per hour to produce refrigeration at the rate of one ton per day should be determined for the conditions "from and at" the boiling point of the respective liquids corresponding to standard atmospheric pressure.

(2) *General Conditions.*

Evaporation of a liquid from any other temperature into saturated vapor at any other pressure to be reduced to an equivalent evaporation under standard "from and at" conditions by the use of appropriate "from and at" factors.

3. *Nominal or Builder's Rating for Compressors.*

To be made on the basis of a *displacement* under *standard conditions* of a sufficient number of cubic feet of saturated gas per unit of time to represent the absorption of 288,000 British thermal units of heat per twenty-four hours.

(1) *Actual Displacement.*

Since the volume of vapors effectively displaced is less than the volume swept out by the piston, and since this ratio varies with different machines and with the same machine under different conditions, the use of this ratio as a "factor of efficiency"—experimentally determined—would seem advisable.

(2) *Standard Conditions.*

Piston speeds in feet per minute for the various sized machines determined from a regular curve of the average piston speeds employed by representative builders of refrigerating machines should be adopted as a basis on which to determine the nominal rating of a compressor.

Since unit volumes of vapor under different pressures represent different amounts of refrigeration, the amount of gas equivalent to the refrigerating rate of one ton per twenty-four hours should be determined under standard conditions corresponding as closely as possible with average practice. A liquid temperature of 90 degrees Fahr. and a gas temperature of 0 degree Fahr. corresponding to a gauge back pressure of 15.67 pounds has already been suggested by the "Standard Ton" Committee.

(3) *General Conditions.*

The equivalent performance of the machine when working under other than standard conditions should be brought to the standard basis by the use of appropriate factors.

Refrigerating and Steam Systems Compared.

2. Thermodynamically the boiler of a steam plant, composed of a boiler, an engine and a surface condenser, corresponds very closely to the cooler (boiler) of a refrigeration plant composed of a cooler, a compressor—or generator—and a condenser. A shell boiler is duplicated in the cooler of the shell type of construction and a water-tube boiler in direct expansion coils. Each is a receptacle for the evaporation of its respective working medium; this evaporation being accomplished in the former case by heat liberated in the combustion of coal, and in the latter, by animal or solar heat with which the liquid comes in contact in its journey through expansion pipes immersed in comparatively hot brine or the atmosphere of cold storage rooms.

Actual Performance.

3. The actual capacity of each system is directly dependent upon the amount of the working liquid evaporated per unit of time, from a given temperature, and at a given pressure and corresponding temperature, but the ability of either system to attain a given capacity, is dependent on physical laws involved in the construction and operation of the system.

British Thermal Units Basis of Boiler Horse-Power (already established).

4. The unit of steam boiler capacity or boiler horse-power has been fixed by this society as equivalent to the evaporation of 30 pounds of water from a feed water temperature of 100 degrees Fahr. into steam at 70 pounds gauge pressure. Such a unit is equivalent to 33,306 British thermal units of heat or 34.5 units of evaporation, the latter being the amount of heat necessary to evaporate one pound of water "from and at" 212 degrees Fahr. (from feed water at 212 degrees Fahr. into steam at atmospheric pressure). For convenience in reference, equivalents of the various units employed in boiler horse-power computations are given below in Table I.

TABLE I.

FUNDAMENTAL DATA OF BOILER HORSE-POWER.

Pounds Water per Hour.	Temperature, F°.	Gauge Pressure.	B.T.U's.	Units of Evaporation.	Boiler H. P.
1	212	0	965.7	1.	.02898
34.5	212	0	33,316.	34.5	1.
1	100	70	1,110.2	1.1497	.033322
30	100	70	33,306.	34.488	.99965

FUNDAMENTAL DATA OF STANDARD TON OF REFRIGERATION.

TIME.	Cubic Feet.	Pounds.	Temp., F°.	Gauge Pressure.	B.T.U's.	Units of Evap.	Tons of Refrig.
.....	18.	1.	$-28\frac{1}{2}$	0	572.78	1.	.001988
Per 24 hrs..	9,050.58	502.81	$-28\frac{1}{2}$	0	288,000.	502.81	1.
Per hour...	377.10	20.950	$-28\frac{1}{2}$	0	12,000.	20.951	1.
Per minute.	6.285	.349	$-28\frac{1}{2}$	0	200.	.349	1.
....	9.028	1.	90°	15.67 (0°F.)	$\{ 555.5 - 111.1 \} = 444.4$	.7758	.001543
Per 24 hrs..	585.072	648.064	90°	15.67 (0°F.)	288,000.	502.81	1.
Per hour...	243.78	27.002	90°	15.67 (0°F.)	12,000.	20.951	1.
Per minute.	4.063	4.5	90°	15.67 (0°F.)	200.	.349	1.

British Thermal Units Basis of Standard Ton of Refrigeration (proposed).

5. The number of negative British thermal units in a pound of ice melted from and at 32 degrees Fahr. being taken at 144, the heat units per ton of 2,000 pounds is 288,000, which value

has been recommended by the committee appointed by this society, as a basis for a *Standard Ton of Refrigeration.*

6. There being twenty-four hours in a day, the number of negative British thermal units per hour to equal the rate of one ton of refrigeration per twenty-four hours is $\frac{288,000}{24} = 1,200 = K$ (1); and there being 1,440 minutes in a day the number of negative British thermal units per minute to equal the rate of one ton of refrigeration per twenty-four hours is $\frac{288,000}{1,440} = 200.$ (2).

7. The number of pounds of any given refrigerating medium to be evaporated per hour to produce refrigeration at the rate of one ton per twenty-four hours is obviously the constant $K = 1,200$ divided by the cold producing effect of one pound of the liquid.

8. In general the process of mechanical refrigeration consists of utilizing the heat to be removed from the cooled space, to evaporate some volatile liquid such as anhydrous ammonia, carbon dioxide, etc., at the comparatively low temperatures at which it can be made to boil at convenient pressures, and in subsequent manipulation to raise the potential of this heat to such an altitude that a large per cent. of it is able to gravitate away at the higher level. The amount of heat which cannot escape from the working medium, depends upon its specific heat and its absolute temperature. Before useful work of refrigeration can be done there must first be evaporated a sufficient amount of the liquid to chill the remainder down to the temperature of the cooler, and the amount of negative heat required for this process is the difference in the heat of the liquid under the two conditions, and is denoted by $(Q_{HP} - Q_{BP})$. The amount of refrigeration available for useful work per pound of liquid is obviously the latent heat of volatilization minus the refrigeration exerted on the liquid itself, and is numerically equal to $R_{BP} - (Q_{HP} - Q_{BP})$, where R_{BP} is the latent heat of the liquid at pressure $_{BP}$. Similarly the number of pounds of the working medium to be evaporated per hour from liquid in the condition $_{HP}$ into saturated vapor under the condition $_{BP}$ to produce one ton of refrigeration per twenty-four hours is

$$P = \frac{K}{R_{BP} - (Q_{HP} - Q_{BP})} \quad (3)$$

"From and at" Basis for Steam.

In the case of computations having to do with steam boiler capacity, the complications arising from heating the feed water up to the temperature of the steam within the boiler, and the fact that the latent heat of volatilization varies with every change in pressure are avoided by the use of the "from and at 212 degree" standard. Equation three given above can be greatly simplified by pursuing a similar course. If the liquid is already at the temperature at which it is to be evaporated ($Q_{HP} - Q_{BP}$) becomes zero, which means that the whole refrigerating effect due to the latent heat of volatilization of the liquid will be exerted in useful work, in which case the final form of the equation (3) is

$$P = \frac{K}{R_{BP}}.$$

Working Medium.

9. Up to this point the refrigerating medium has not been defined, but since no two liquids have the same properties, further computations will have to be based on properties peculiar to some given liquid or liquids. As anhydrous ammonia is the medium with which the profession is most familiar, this paper will deal exclusively with that liquid, it being understood, however, that similar calculations might just as well have been made for any other liquid.

Standard Conditions.

10. A temperature of 90 degrees for the liquefaction of the ammonia gas in the condenser and 0 degrees Fahr. for the temperature of evaporation of the liquid ammonia in the cooler, has been suggested by the "standard ton" committee as being the most suitable standard temperatures under which the standard ton of refrigeration should be produced.

Refrigeration Expended on the Liquid.

11. The latent heat of volatilization of anhydrous ammonia at 0 degrees Fahr. is 555.50 British thermal units. *Assuming* its

specific heat to be 1.2345 the amount of refrigeration necessary to cool the liquid from 90 degrees Fahr. to 0 degrees Fahr. will be $(Q_{HP} - Q_{BP}) = (T_{HP} - T_{BP}) S = 90 \times 1.2345 = 111.10$ and $555.50 - 111.10 = 444.40$, the net number of British thermal units of negative heat available in a pound of liquid evaporated under standard conditions.* On this basis the evaporation of 27 pounds of ammonia per hour, or 648 pounds per day, would be equivalent to one ton of refrigeration per twenty-four hours.

Factors of Equivalent Evaporation.

12. In the case of steam the factor of equivalent evaporation from and at 212 degrees Fahr. is the ratio of the amount of heat required to evaporate a pound of water from any given feed water temperature T, into steam at any given gauge pressure P, to 965.7 (the number of British thermal units in a unit of evaporation), the latter being the latent heat of steam at 212 degrees Fahr. By multiplying the amount of water evaporated under any given set of conditions by the proper factor of equivalent evaporation, we at once arrive at the amount of water which would have been evaporated had the same amount of heat been expended in evaporating it under the standard conditions of

*A standard unit of refrigeration, such as this Society is seeking to establish, should first of all be a *commercial unit* and need not necessarily stand for *absolute* physical and mathematical accuracy. While agreeing very closely with the average of several determinations, 144 British thermal units, for example, can hardly be expected to be the *absolutely* correct value for the latent heat of ice, but the advantage which the engineer finds in the fact that with this value, the rate of one ton of refrigeration or 288,000 British thermal units per 24 hours of 1440 minutes, corresponds exactly with 12,000 British thermal units per hour and 200 British thermal units per minute, is of far more commercial value than absolute accuracy bought at the price of more unwieldy whole numbers with long decimal appendages.

The value 1.2345, used as the specific heat of liquid anhydrous ammonia (paragraph 11), gives peculiarly simple and easily remembered results. The number itself is composed of the first five numerals, which when multiplied by the difference in degrees between the recommended standard condenser and refrigerator temperatures gives 111.1, a number composed of four ones, which subtracted from the latent heat of vaporization for anhydrous ammonia at 0° F—555.5, a number composed of four 5's, gives as the net number of British thermal units available per pound of liquid evaporated under standard conditions 444.4, a number composed of four 4's. The value 1.2345 is a pure assumption, not claimed to correspond with average determinations, but, being approximately accurate, is introduced on account of the advantages offered by the strikingly simple results entailed by its use.

"from and at 212 degrees Fahr." If in the above ratio, instead of using 965.7 (the number of British thermal units, required to evaporate a pound of water from and at 212 degrees), 1110.2 (the number of British thermal units necessary to evaporate a pound of 100 degrees water into 70 pound steam), had been used, the resulting quotients would have been *factors of equivalent evaporation from* 100 *degrees and at* 70 *pounds pressure.* Complete tables of both these factors have been calculated, and while 100 degree water, evaporated into 70 pound steam, more nearly approaches practice than 212 degree water evaporated at atmospheric pressure, yet practice has voted almost unanimously in favor of the 212 degree standard.

"*From and at*" *Basis for Ammonia.*

13. In practice we often find refrigerating liquids being evaporated at atmospheric pressure, a condition which has comparatively little application in the case of steam. Atmospheric pressure and corresponding temperature may, therefore, be used even more rationally as standard conditions in the case of refrigeration, than in that of steam. Under such conditions equation (3)

$$P = \frac{K}{R_{BP} - (Q_{HP} - Q_{BP})}$$

when applied to ammonia becomes

$$P = \frac{K}{R_{-28}} = \frac{1,200}{572.78} = 20.950,$$

which is the number of pounds of ammonia per hour evaporated "from and at" minus 28½ degrees Fahr. and atmospheric pressure, to be equivalent to one ton of refrigeration per twenty-four hours. Multiplying both numerator and denominator of the right-hand member of equation (3) by R_{-28} gives—

$$P = \frac{K}{R_{-28}} \times \frac{R_{-28}}{R_{BP} - (Q_{HP} - Q_{BP})}$$

the first term of which we recognize as equation (3) applied to evaporation from and at — 28½ degrees Fahr. and atmospheric pressure, and the second term of which, is the ratio of the amount of heat required to evaporate one pound of anhydrous ammonia from and at — 28½ degrees Fahr. and atmospheric pressure, to that required to evaporate it from the liquid condition, HP into gas

of a pressure BP, and is therefore an expression for F, a factor of equivalent evaporation for ammonia from and at — $28\frac{1}{2}$ degrees Fahr.

Example.

14. Assuming the recommended standard conditions of 90 degrees Fahr. for the temperature of liquefaction, and 0 degrees Fahr. for the temperature of volatilization and substituting in the expression for the factor,

$$\frac{R_{-28}}{R_{BP} - (Q_{HP} - Q_{BP})} \text{ becomes } \frac{572.78}{555.50 - 111.10} = 1.2888,$$

which is a factor of evaporation from and at — $28\frac{1}{2}$ degrees for the standard conditions and means that the refrigeration produced by the evaporation of one pound of ammonia from and at $28\frac{1}{2}$ degrees, would be 1.2888 times as much as when evaporated from liquid at 90 degrees Fahr. into vapor at 0 degrees Fahr. (15.67 pounds gauge pressure), and conversely if, as has been shown above, 20.950 pounds of ammonia must be evaporated per hour from and at — $28\frac{1}{2}$ degrees Fahr. to produce a ton of refrigeration per 24 hours, $1.2888 \times 20.950 = 27{,}000$ pounds, as already determined, must be evaporated from 90 degrees liquid into gas at 15.67 pounds gauge pressure (0 degrees Fahr.).

15. Further numerical substitutions, of Q and R values corresponding to any given conditions of temperature and pressure (these taken from tables of properties of saturated ammonia), give corresponding factors of equivalent evaporation a number of which, covering practical commercial conditions have been calculated and are given in Table II. (Factor A.)

16. Numerical substitution in equation (3) gives the number of pounds of anhydrous ammonia to be evaporated per hour from a liquid temperature HP into a saturated gas at any pressure PB to produce refrigeration at the rate of one ton per twenty-four hours. These factors (B, Table II.), are figured on the basis that 288,000 British thermal units is the equivalent of a ton of refrigeration, but owing to the lack of agreement among authorities regarding the value of the specific heat of anhydrous ammonia, *its value has been taken as unity.* Factors C, Table II., are the volumes in cubic feet per minute of saturated gas which must pass through the compressor to produce a ton of refrigeration per twenty-four hours.

TABLE II.

A—Factors of equivalent evaporation for NH_3 " from and at " $-28\frac{1}{2}$° F.

B—Pounds of ammonia to be evaporated per minute at various back pressures from liquid at various temperatures to equal the rate of one ton of refrigeration per 24 hours.

C—Cubic feet of ammonia gas to be handled by compressor per minute to produce refrigeration at the rate of one ton per 24 hours.

$$\textbf{A, Factor} = \frac{R - 28}{R_{BP} - (Q_{HP} - Q_{BP})} \qquad \textbf{B, Pounds of ammonia} = \frac{144 \times 2000}{1440\, R_{BP} - (Q_{HP} - Q_{BP})}$$

(Data from De V. Wood's Ammonia Tables.—Computed by F. E. Matthews, Refrigerating Eng.

HP. Head Pressure, Condenser or Gauge Pressure and Corresponding Temperature.

l / *w* / *B.P.*		100 lb. 63.5°	110 lb. 68°	120 lb. 72.6°	130 lb. 77.4°	140 lb. 80.3°	150 lb. 83.8°	160 lb. 87.4°	170 lb. 90.8°	180 lb. 93.8°	190 lb. 96.3°	200 lb. 100°	Temp. F.°
572.78	A	1.1913	1.2026	1.2145	1.2268	1.2347	1.2441	1.2533	1.2625	1.2716	1.2804	1.2892	−28.5°
.0556	B	.4159	.4199	.4240	.4284	.4310	.4343	.4376	.4408	.4440	.4470	.4501	
0	C	7.482	7.551	7.626	7.703	7.761	7.812	7.870	7.929	7.986	8.041	8.095	
566.14	A	1.1806	1.1917	1.2034	1.2154	1.2231	1.2341	1.2417	1.2502	1.2594	1.2680	1.2767	−17.5°
.0133	B	.4122	.4160	.4202	.4243	.4271	.4308	.4335	.4366	.4397	.4437	.4458	
5	C	5.636	5.675	5.732	5.790	5.836	5.878	5.914	5.970	5.999	6.039	6.081	
560.69	A	1.1721	1.1829	1.1944	1.2038	1.2137	1.2231	1.2322	1.2409	1.2497	1.2581	1.2667	− 8.5°
.0910	B	.4093	.4130	.4171	.4204	.4237	.4271	.4302	.4332	.4363	.4392	.4423	
10	C	4.502	4.543	4.587	4.625	4.662	4.698	4.733	4.766	4.799	4.833	4.865	
556.11	A	1.1651	1.1758	1.1871	1.1990	1.2062	1.2155	1.2245	1.2334	1.2417	1.2501	1.2585	− 1.°
.1083	B	.4068	.4106	.4145	.4186	.4211	.4244	.4276	.4288	.4336	.4365	.4394	
15	C	3.756	3.791	3.827	3.866	3.889	3.918	3.948	3.975	4.003	4.030	4.058	
552.83	A	1.1571	1.1677	1.1789	1.1906	1.1977	1.2068	1.2168	1.2242	1.2327	1.2409	1.2499	5.66°
.1258	B	.4040	.4077	.4116	.4158	.4182	.4214	.4245	.4275	.4304	.4333	.4362	
20	C	3.211	3.241	3.272	3.305	3.324	3.350	3.375	3.398	3.422	3.444	3.467	
548.40	A	1.1520	1.1631	1.1746	1.1801	1.1933	1.2023	1.2112	1.2195	1.2280	1.2361	1.2443	11.5°
.1429	B	.4025	.4062	.4102	.4140	.4167	.4198	.4229	.4258	.4287	.4316	.4345	
25	C	2.819	2.843	2.870	2.898	2.916	2.938	2.959	2.980	3.000	3.020	3.040	
545.13	A	1.1491	1.1596	1.1706	1.1821	1.1891	1.1983	1.2068	1.2152	1.2237	1.2318	1.2399	16.8°
.1600	B	.4013	.4049	.4088	.4128	.4152	.4184	.4213	.4243	.4273	.4300	.4329	
30	C	2.507	2.530	2.555	2.580	2.600	2.615	2.633	2.653	2.671	2.687	2.706	
542.80	A	1.1431	1.1535	1.1644	1.1758	1.1827	1.1916	1.2003	1.2085	1.2168	1.2249	1.2330	21.7°
.1766	B	.3991	.4028	.4066	.4105	.4130	.4161	.4188	.4220	.4249	.4277	.4305	
35	C	2.260	2.280	2.302	2.325	2.338	2.356	2.373	2.390	2.406	2.422	2.443	
539.35	A	1.1411	1.1514	1.1623	1.1736	1.1805	1.1894	1.1981	1.2062	1.2145	1.2225	1.2303	26.1°
.1941	B	.3984	.4020	.4058	.4098	.4122	.4153	.4183	.4211	.4240	.4269	.4296	
40	C	2.052	2.071	2.090	2.111	2.123	2.139	2.155	2.175	2.185	2.300	2.214	

l Latent Heat of Volatilization, *Lbp.* *w* Weight of Vapor per Cubic Foot. *BP.* Back Pressure, Suction or Gauge Pressure.

(Data from De V. Wood's ammonia tables recalculated by Geo. Davidson, M.E.—Compend of Mechanical Refrigeration.)

Proposed Equivalent Standard Units.

17. On a basis similar to that of the present boiler horsepower of 30 pounds of water evaporated per hour from feed water at a temperature of 100 degrees Fahr. into saturated steam at 70 pounds gauge pressure, which requires 33,306 British thermal units of heat or 34.5 units of evaporation, each of which is equal to 965.7 British thermal units—the amount of heat re-

quired to evaporate one pound of water from and at 212 degrees and atmospheric pressure—may be established a *Standard Ton of Refrigeration, equivalent to* 27 *pounds of anhydrous ammonia evaporated per hour from liquid at a temperature of* 90 *degrees Fahr. into saturated vapor at* 15.67 *pounds gauge pressure* (0 *degrees Fahr.*), *which requires* 12,000 *British thermal units of heat or* 20.950 *units of evaporation, each of which is equal to* 572.78 *British thermal units—the amount of heat required to evaporate one pound of ammonia from a temperature of* $-28\frac{1}{2}$ *degrees into saturated vapor at atmospheric pressure.*

Nominal Rating.

18. While the equivalent of the evaporation of 30 pounds of water and 27 pounds of anhydrous ammonia under their respective standard conditions furnish convenient numerical means of arriving at the actual performance of a steam boiler and a refrigerating machine respectively, when they are in operation and the amount of the working medium can be measured, commercial practice demands for both systems a means of calculating, at least approximately, their capacity when not in operation or when the exact amount of the working fluid actually evaporated cannot be directly determined. Such a demand in the case of the steam boiler has led to the establishing of square-foot ratings based on the physical ability of the boilers of the various types to produce standard thermal units of boiler capacity, these square-foot standards being used to determine what is known as the nominal or builders' rating. Had the question of "Standard Unit of Refrigeration" been likewise considered independently from that of nominal or builders' rating, much of the chaos resulting from confusion of the two subjects might have been avoided.

Metering Vapors of Water and Ammonia.

19. Unfortunately the amount of ammonia evaporated in a refrigerator cannot be so easily determined as the amount of water evaporated in a steam boiler. If, for any reason, however, we were unable to determine the performance of the boiler by the amount of water going into it, we would naturally try to determine its capacity from the amount of steam coming out of it. For this purpose we might employ a steam meter and if this proved unreliable we would probably compute the steam consump-

tion of the various engines drawing their supply from the boiler in question. This we would accomplish by means of a series of carefully taken steam indicator cards. A similar course is open in the case of compression refrigerating systems with the advantage that intelligently interpreted ammonia cards are far more accurate than steam cards, as an index to the number of pounds of the working medium which passes the cylinder. As very few plants have ever been equipped with meters for measuring the amount of refrigerating liquid, and as the accuracy of such instruments when applied to liquids often in a boiling condition, may well be questioned, the use of the compressor as a meter, after it has been carefully callibrated by frequent indicator cards seems to be the only alternative.

20. Were it possible to construct a compressor, the effective cubical gas displacement of which, at the cooler back pressure, would equal the volume swept out by the piston, there would be no use for the indicator card, and, similar to the square-foot ratings laid down in the case of steam boilers, a standard unit of nominal refrigerating capacity could be established on the simple basis of cubical compressor displacement in the case of the compression machine and cubical ammonia pump displacement coupled with factors involving the strengths of the ammonia liquors in the case of the absorption plant.

21. In practice, however, slippage in the ammonia pump entails an error of uncertain magnitude, and difficulties no less formidable arise in the case of the compressor. On account of re-expansion of high pressure gas filling clearance spaces and the resistance which the pipes, valves and port openings offer to the passage of the low pressure gas entering the compressor cylinder, the suction valves do not open to admit the fresh complement of gas until the piston has advanced some distance from the end of stroke, and the pressure of the gas within the cylinder is often appreciably less than that in the cooler. The existence of these conditions demands that, in connection with the apparent cubical displacement of compressors there be employed "factors of displacement efficiency" based on actual observations of the performance of the various machines.

Effective Displacement.

22. In the past, general practice has been to combine the loss due to imperfect cylinder fillment with that due to the refrigera-

tion that has to be expended on the liquid itself in order to cool it down to the temperature of the cooler, and call the whole 25 per cent. of the apparent capacity, thus erroneously assuming the efficiency of all systems to be 75 per cent. under all conditions. It is interesting to note that the efficiency shown by the indicator diagram analyzed below, saying nothing about cooling the liquid and other lesser inherent losses, is little above the conventional 75 per cent., and this under conditions of speed and condenser pressure most conducive to high economy.

23. The indicator diagram shown in Fig. 192 is a reproduction of an actual card taken from an old style 13 by 38-inch horizontal machine running 45 revolutions per minute. This card was selected on account of the great re-expansion that it shows, which while the head pressure is only about six atmospheres, and the back pressure is about one and one-half atmospheres (conditions very favorable for compressors of such faulty design), the effect of re-expansion is sufficient to keep the suction valves closed through the first 11.4 per cent. of the stroke.

Calculating Clearance.

24. While the example given in the following paragraph is somewhat of a digression, the method employed in calculating the clearance based on the assumption that the re-expansion is adiabatic may be useful in working up indicator cards which show a considerable re-expansion.

25. If P and V (Fig. 192) denote absolute pressure and volume at point A, and $P_1 V_1$ denote similar absolute pressure and volume at B, the equation of an adiabatic through these two points will be $P V^{1.3} = P_1 V_1^{1.3}$. V at A, however, is the actual unknown clearance, which may be denoted by x, and V_1 at B is the actual clearance x plus re-expansion $= x + 11.4$ per cent. Substituting these values as well as those for the absolute pressures in the equation of the adiabatic, $P V^{1.3} = P_1 V_1^{1.3}$ becomes $(72 + 15)x^{1.3} = (4.78 + 15)\,(11.4 + x)^{1.3}$, which by dividing both sides of the equation by $19.78x^{1.3}$ becomes $\frac{87}{19.78} = \left(\frac{11.4 + x}{x}\right)^{1.3}$, and by raising both sides of the equation to the -1.3 power we have $\left(\frac{87}{19.78}\right)^{.769} = \frac{11.4 + x}{x}$. By logs. $\left(\frac{87}{19.78}\right)^{.769} = 3.123 = \frac{11.4 + x}{x}$, from which $x = 5.370$, the per cent. actual clearance, which can be laid off to scale at the right of the diagram, Fig. 192.

Factor of Efficiency.

26. As a simple engine room method of determining the effective displacement of a compressor, which is virtually the ratio of the actual to the apparent weights of ammonia handled by the machine, and may be taken as a "factor of efficiency," the following method is proposed.

27. For all practical purposes the weight of ammonia gas may

GRAPHICAL METHOD FOR DETERMINING FACTOR OF EFFICIENCY OF A COMPRESSOR

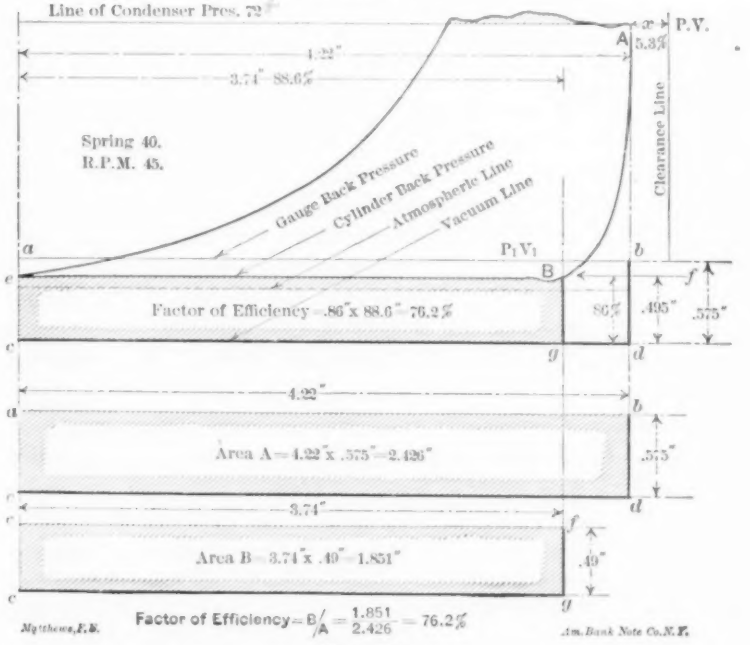

FIG. 192.

be said to be proportioned to its absolute pressure, and within narrow limits the amount of refrigeration represented in a cubic foot of ammonia gas must, therefore, likewise be proportional to the absolute pressure. In Fig. 192, lines *ab*, *cd* and *ef*, represent the lines of actual back pressure, absolute vacuum, and back pressure within the cylinder respectively, the actual absolute back pressure being 23 pounds per square inch and the back pressure within the cylinder being 19.78 pounds, or only 86 per cent. as great.

The weight of the ammonia gas per cubic foot within the cylinder is, in accordance with the foregoing assumptions, only 86 per cent. of that in the suction line outside the compressor valves. The point *B* at which the suction valve opens to admit cold gas to the cylinder is found to be 11.4 per cent. of the length of the cylinder from the end of the stroke, and since at this point the cylinder is filled with hot gas at the pressure of the incoming cold gas, the per cent. volume of the cylinder remaining to be filled by the incoming gas will be reduced to 88.6 per cent. Combining the fact that the compressor is taking in only 88.6 per cent. as many cubic feet of gas as is swept out by the piston, with the fact that each cubic foot of the gas effectively displaced weighs only 86 per cent. as much as we would expect from the actual observed back pressure, would give as a "factor of efficiency," of the compressor $0.86 \times 88.6 = 76.19$ per cent. Graphically the problem lends itself to a very simple solution. The *apparent* displacement of the compressor in pounds is proportional to *A* the area of the rectangle *abdc*—in which *ab* represents the length of the cylinder and is proportional to the volume swept out by the piston per stroke, and *bd* represents the absolute back pressure and is proportional to the weight of the ammonia gas per cubic foot in the cooler. Similarly the *actual* displacement of the compressor in pounds, is proportional to *B* the area of the rectangle *efgc*, in which *ef* represents the volume of cold gas taken into the cylinder and *fg* represents the weight of gas per cubic foot within the cylinder. The ratio (B/A) of the areas of these two rectangles, which may be computed from their lineal dimensions measured with an ordinary scale and expressed in any units whatsoever, gives the per cent. effective displacement of the compressor. Although this factor does not take into account cylinder heating and other lesser sources of inefficiency inherent to the compressor, it does account fairly accurately for the principal ones, and best of all it involves no lengthy mathematical process, but on the other hand can be employed by any one who can measure accurately with a scale and can multiply and divide. To be mathematically correct the factor should be corrected for some standard head and back pressure such as already suggested, viz: 168 and 15.67 pounds corresponding to 90 degrees and 0 degrees Fahr. respectively. Since the error arising from the use of the uncorrected factor of displacement efficiency varies as the difference between the observed and the standard conditions and, as the lat-

ter are supposed to correspond with the average of the former, the error introduced by using the uncorrected factor can, in the majority of cases, be brought within the error of the observations by slightly manipulating the pressures while the cards are being taken. Under these conditions, as has been shown above, 27 pounds of ammonia must be evaporated per hour to produce one ton of refrigeration per twenty-four hours. The volume per pound of saturated gas at 15.67 pounds pressure being 9.028 cubic feet, 243.75 cubic feet per hour or 4.06 cubic feet per minute must be passed through the compressor for each ton produced.

28. Since the number of cubic feet of piston displacement of a given compressor varies with its factor of efficiency, various types of compressors should be given factors according to their efficiency in handling gas just as various types of boilers are given square-foot ratings according to the efficiency of their heating surfaces in transmitting heat. The following simple example, in which is calculated the capacity, under the proposed standard conditions, of the compressor from which the indicator diagram (Fig. 192) was taken, will suffice to illustrate the application of the method.

By interpolation from Table II the number of pounds of ammonia per minute from 90 deg. liquid to 0 deg. gas to equal the rate of one ton of refrigeration per twenty-four hours, is found to be ..... .4273

At this pressure the corresponding volume is ..... 3.857 cu. ft.

Piston speed in feet per minute of a 38-inch stroke machine running 45 r. p. m. (See Table III) ..... = 284.

Volume in cu. ft. swept out by piston per foot of travel. (Table IV) ..... = 1.969

Volume in cu. ft. swept out by piston per minute = 284 feet of travel ..... = 559.19

Per cent. of effective displacement of compressor determined by graphical method shown in Fig. 192. ..... = 76.19

Effective gas displacement in cu. ft. per minute $76.19 \times 559.19 = 426.047$

Capacity in tons per twenty-four hours under standard conditions, $\frac{426.047}{3.857}$ ..... $= 110.4$

29. The equivalent performance under any other conditions such as 300 feet per minute piston speed, 150 pounds head pressure and 15 pounds back pressure will be proportional to the direct ratio of the piston speed and the inverse ratio of the volumes of gas, as given in Table II., required per minute per ton. Numerically this would be—

$$110.4 \times \frac{300}{284} \times \frac{3.857}{3.918} = 114.8 \text{ tons.}$$

TABLE III.—Piston Speed in Feet per Minute.

Calculated by F. E. Matthews

Length of Stroke in In.	Revolutions per Minute.																									
	1	24	26	28	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	
1	.1666	4	4.33	4.66	5	5.16	5.33	5.5	5.66	5.83	6	6.16	6.33	6.5	6.66	6.83	7	7.16	7.33	7.5	7.66	7.83	8	8.16	8.33	
2	.333	8	8.66	9.33	10	10.33	10.66	11	11.33	11.66	12	12.33	12.66	13	13.33	13.66	14	14.33	14.66	15	15.33	15.66	16	16.83	16.66	
4	.666	16	17.33	18.66	20	20.66	21.33	22	22.66	23.33	24	24.66	25.33	26	26.66	27.33	28	28.66	29.33	30	30.66	31.33	32	32.66	33.33	
6	1	24	26	28	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	
8	1.333	32	34.66	37.33	40	41.33	42.66	44	45.33	46.66	48	49.33	50.66	52	53.33	54.66	56	57.33	58.66	60	61.33	62.66	64	65.33	66.66	
9	1.5	36	39	42	45	46.56	48	49.5	51	52.5	54	55.5	57	58.5	60	61.5	63	64.5	66	67.5	69	70.5	72	73.5	75	
10	1.666	40	43.33	46.66	50	51.66	53.33	55	56.66	58.33	60	61.66	63.33	65	66.66	68.33	70	71.66	73.33	75	76.66	78.33	80	81.66	83.33	
11	1.833	44	47.66	51.33	55	56.83	58.66	60.5	62.33	64.16	66	67.83	69.66	71.5	73.33	75.16	77	78.83	80.66	82.5	84.33	86.16	88	89.83	91.66	
12	2	48	52	56	60	62	64	66	68	70	72	74	76	78	80	82	84	86	88	90	92	94	96	98	100	
13	2.166	52	56.33	60.66	65	67.16	69.33	71.5	73.66	75.83	78	80.16	82.33	84.5	86.66	88.83	91	93.16	95.33	97.5	99.66	101.83	104	106.16	108.33	
14	2.333	56	60.66	65.33	70	72.33	74.66	77	79.33	81.66	84	86.33	88.66	91	93.33	95.66	98	100.33	102.66	105	107.33	109.66	112	114.33	116.66	
15	2.5	60	65	70	75	77.5	80	82.5	85	87.5	90	92.5	95	97.5	100	102.5	105	107.5	110	112.5	115	117.5	120	122.5	125	
16	2.666	64	69.33	74.66	80	82.66	85.33	88	90.66	93.33	96	98.66	101.33	104	106.66	109.33	112	114.66	117.33	120	122.66	125.33	128	130.66	133.33	
17	2.833	68	73.66	79.33	85	87.83	90.66	93.5	96.33	99.16	102	104.83	107.66	110.5	113.33	116.16	119	121.83	124.66	127.5	130.33	133.16	136	138.83	141.66	
18	3	72	78	84	90	93	96	99	102	105	108	111	114	117	120	123	126	129	132	135	138	141	144	147	150	
19	3.166	76	82.33	88.66	95	98.16	101.33	104.5	107.66	110.83	114	117.16	120.33	123.5	126.66	129.83	133	136.16	139.33	142.5	145.66	148.83	152	155.16	158.33	
20	3.333	80	86.66	93.33	100	103.33	106.66	110	113.33	116.66	120	123.33	126.66	130	133.33	136.66	140	143.33	146.66	150	153.33	156.66	160	163.33	166.66	
22	3.666	88	95.33	102.66	110	113.66	117.33	121	124.66	128.33	132	135.66	139.33	143	146.66	150.33	154	157.66	161.33	165	168.66	172.33	176	179.66	183.33	
24	4	96	104	112	120	124	128	132	136	140	144	148	152	156	160	164	168	172	176	180	184	188	192	196	200	
26	4.333	104	112.66	121.33	130	134.33	138.66	143	147.33	151.66	156	160.33	164.66	169	173.33	177.66	182	186.33	190.66	195	199.33	203.66	208	212.33	216.66	
28	4.666	112	121.33	130.66	140	144.66	149.33	154	158.66	163.33	168	172.66	177.33	182	186.66	191.33	196	200.66	205.33	210	214.66	219.33	224	228.66	233.33	
30	5	120	130	140	150	155	160	165	170	175	180	185	190	195	200	205	210	215	220	225	230	235	240	245	250	
32	5.333	128	138.66	149.33	160	165.33	170.66	176	181.33	186.66	192	197.33	202.66	208	213.33	218.66	224	229.33	234.66	240	245.33	250.66	256	261.33	266.66	
34	5.666	136	147.33	158.66	170	175.66	181.33	187	192.66	198.33	204	209.66	215.33	221	226.66	232.33	238	243.66	249.33	255	260.66	266.33	272	277.66	283.33	
36	6	144	156	168	180	186	192	196	204	210	216	222	228	234	240	246	252	258	264	270	276	282	288	294	300	
38	6.333	152	164.66	177.33	190	196.33	202.66	209	215.33	221.66	228	234.33	240.66	241	253.33	259.66	266	272.33	278.66	285	291.33	297.66	304	310.33	316.66	
40	6.666	160	173.33	186.66	200	206.66	213.33	220	226.66	233.33	240	246.66	253.33	260	266.66	273.33	280	286.66	293.33	300	306.66	313.33	320	326.66	333.33	
42	7	168	182	196	210	217	224	231	238	245	252	259	266	273	280	287	294	301	308	315	322	329	336	343	350	
44	7.333	176	190.66	205.33	220	227.33	234.66	242	249.33	256.66	264	271.33	278.66	286	293.33	300.66	308	315.33	322.66	330	337.33	344.66	352	359.33	366.66	
46	7.666	184	199.33	214.66	230	237.66	245.33	253	260.66	268.33	276	283.66	291.33	299	306.66	314.33	322	329.66	337.33	345	352.66	360.33	368	375.66	383.33	
48	8	192	208	224	240	248	256	264	272	280	288	296	304	312	320	328	336	344	352	360	368	376	384	392	400	
50	8.333	200	216.66	233.33	250	258.33	266.66	275	283.33	291.66	300	308.33	316.66	325	333.33	341.66	350	358.33	366.66	375	383.33	391.66	400	408.33	416.66	
52	8.666	208	225.33	242.66	260	268.66	277.33	286	294.66	303.33	312	320.66	329.33	338	346.66	355.33	364	372.66	381.33	390	398.66	407.33	416	424.66	433.33	
54	9	216	234	252	270	279	288	297	306	315	324	333	342	351	360	369	378	386	395	405	414	423	432	441	450	
56	9.333	224	242.66	261.33	280	289.33	298.66	308	317.33	326.66	336	345.33	354.66	364	373.33	382.66	392	401.33	410.66	420	429.33	438.66	448	457.33	466.66	
58	9.666	232	251.33	270.66	290	299.66	309.33	319	328.66	338.33	348	357.66	367.33	377	386.66	396.33	406	415.66	425.33	435	444.66	454.33	464	473.66	483.33	
60	10	240	260	280	300	310	320	330	340	350	360	370	380	390	400	410	420	430	440	450	460	470	480	490	500	
62	10.333	248	268.66	289.33	310	320.33	330.66	341	351.33	361.66	372	382.33	392.66	403	413.33	423.66	434	444.33	454.66	465	475.33	485.66	496	506.33	516.66	
64	10.666	256	277.33	298.66	320	330.66	341.33	352	362.66	373.33	384	394.66	405.33	416	426.66	437.33	448	458.66	469.33	480	490.66	501.33	512	522.66	533.33	
66	11	264	286	308	330	341	352	363	374	385	396	407	418	429	440	451	462	473	484	495	506	517	528	539	550	
68	11.333	272	294.66	317.33	340	351.33	362.66	374	385.33	396.66	408	419.33	430.66	442	453.33	464.66	476	487.33	498.66	510	521.33	532.66	544	555.33	566.66	
70	11.666	280	303.33	326.66	350	361.66	373.33	385	396.66	408.33	420	431.66	443.33	455	466.66	478.33	490	501.66	513.33	525	536.66	548.33	560	571.66	583.33	
72	12	288	312	336	360	372	384	396	408	420	432	444	456	468	480	492	504	516	528	540	552	564	576	588	600	

30. No general cubic foot rating will apply accurately to all machines, hence for accurate determinations it would be necessary to employ a "factor of efficiency," a proposition for which the writer has described above. The very nature of such a factor

TABLE IV.

CUBICAL DISPLACEMENT OF COMPRESSORS OF VARIOUS SIZES—CUBIC FEET PER FOOT OF PISTON TRAVEL.—F. E. MATTHEWS.

Diameter of Cylinder in Inches.	Even Inches.	Plus ¼ Inch.	Plus ½ Inch.	Plus ¾ Inch.
1	.00540	.00852	.01224	.01668
2	.02184	.02766	.03408	.04128
3	.04908	.05760	.06684	.07668
4	.08724	.09852	.11040	.12300
5	.13632	.15036	.16500	.18036
6	.19636	.21300	.23040	.24852
7	.26724	.28668	.30684	.32760
8	.34908	.37128	.39408	.41760
9	.44184	.46668	.49224	.51852
10	.54540	.57300	.60132	.53024
11	.66060	.69024	.72132	.75300
12	.78540	.81850	.85224	.88668
13	.92172	.95760	.99396	1.03116
14	1.0689	1.10748	1.14672	1.18556
15	1.2271	1.26840	1.31040	1.35300
16	1.3963	1.44024	1.48488	1.53021
17	1.5762	1.62300	1.67028	1.71840
18	1.7671	1.81650	1.86672	1.91748
19	1.9689	2.02116	2.07396	2.12748
20	2.1817	2.13656	2.29212	2.34840
21	2.4053	2.46288	2.52120	2.58012
22	2.6397	2.70072	2.76120	2.82280
23	2.8852	2.94828	3.01200	3.07644
24	3.1416	3.20736	3.27364	3.34104
25	3.4088	3.47736	3.54666	3.61644
26	3.6870	3.75828	3.83016	3.90276
27	3.9760	4.05000	4.12476	4.20000
28	4.2760	4.35276	4.43016	4.50816
29	4.5870	4.66632	4.74648	4.82524
30	4.9081	4.99092	5.07372	5.15724

would undoubtedly lead to endless controversy among the builders of various types of machines, and a liberal compromise will have to be made along the lines of assumed efficiency. Until such factors can be agreed upon an assumed efficiency of ninety per cent. might be involved in the use of 4.4 cubic feet of gas per minute per ton under the standard conditions. The curves in Fig. 193

TABLE VI.

TABLE SHOWING EFFECT OF PROPOSED BASIS OF RATING ON THE PRESENT RATINGS OF FOUR WELL-KNOWN BUILDERS OF REFRIGERATING MACHINES.—F. E. MATTHEWS.

	BUILDER A.							BUILDER B.							BUILDER C.						BUILDER D.						
Builders' Ton-rating	20	45	90	175	225	300	500	20	40	60	80	150	200	350	25	50	75	150	225	450	25	35	50	100	150	200	400
R. P. M. (Fig. 193) on which rating is based	95	73	67	63	57	53	53	90	70	65	60	50	50	50	70	66	60	57	50	50	56	54	56	52	55	52	43
Piston speed A (Fig. 193) on which rating is based	190	219	268	336	342	371	400	225	233	260	280	300	300	400	198	234	250	285	300	400	204	200	244	286	275	286	286
4.4 cu. ft. per min. Rating, a', corresponding to 90° E standard conditions	19	42	85	166	209	287	446	23.4	41.6	58.8	78	156	188	361	25.8	42.9	69.7	140	214	438	25.5	35.8	51.1	102.3	153.4	204	409
R. P. M.—from curve—(Fig 193). Averages of Builders A, B, C and D	90	70	59	55	52	51	46	90	73	64	60	55	53	52	83	68	61	55	52	48	83	75	68	58	55	53	48
Piston Speed—from curve—"x." Averages of Builders A, B, C and D (Fig. 193).	300	234	276	295	308	340	425	200	228	250	270	291	300	358	208	240	265	291	308	400	208	221	240	280	291	300	378
Rating 4.4 cu. ft. and Piston Speed "x"	20	45.32	87.6	146.2	193.3	264.	476.3	20	40.7	56.5	75.3	151.6	188.2	323.5	27	46.5	73.9	104.4	220	437.2	28.4	39.45	50.32	100.3	174.6	214	539.7

TABLE V.

AVERAGE R. P. M. AND PISTON SPEEDS DETERMINED FROM FIG. 193.

Ton	20	25	30	35	40	45	50	55	60	65	70	75	80	90	100	110	120	130	140	160	180	200	220	240	260	280	300	350	400	450	500
R. P. M "y"	90	83	78	75	73	70	68	66	64	63	62	61	60	59	58½	58	57	56½	56	55	54	53	52½	52	51½	51½	51	49½	48½	47½	46
Piston Speeds "x"	200	208	215	221	228	234	240	246	250	256	261	265	270	276	280	283	286	288	290	293	296	300	306	315	323	332	340	358	378	400	425

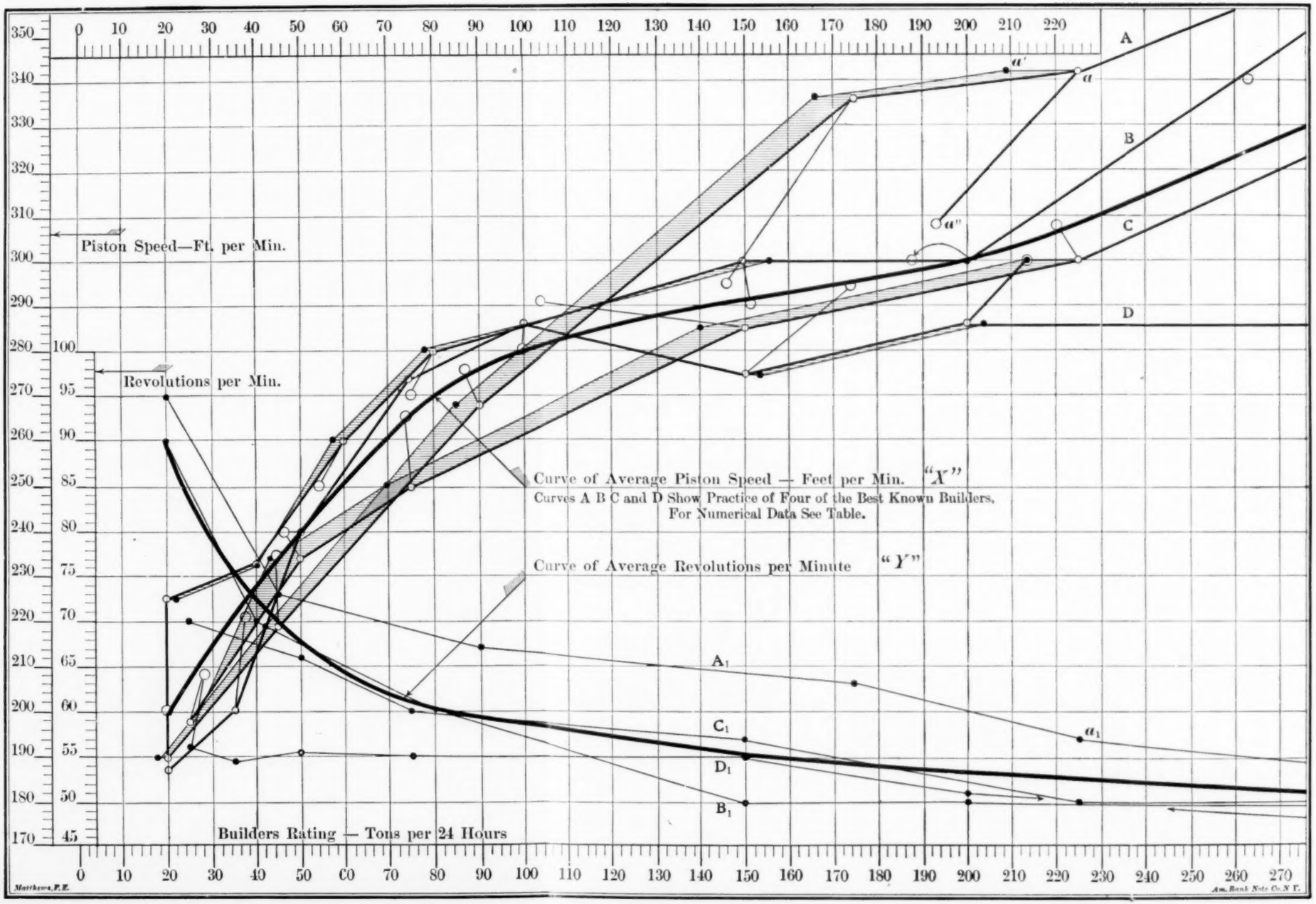
Piston Speed—Ft. per Min.
Revolutions per Min.
Builders Rating — Tons per 24 Hours
Curve of Average Piston Speed — Feet per Min. "X"
Curves A B C and D Show Practice of Four of the Best Known Builders.
For Numerical Data See Table.
Curve of Average Revolutions per Minute "Y"
A
B
C
D
a'
a
a''
A1
C1
D1
B1
a1
Matthews, P.E.
Am. Bank Note Co. N.Y.

FIG. 193.

show how nearly 4.4 cubic feet per ton corresponds to the ratings employed by four of the best known builders of refrigerating machinery.

31. The most difficult factor encountered in establishing a basis for a nominal or builders' rating is that of standard speed. Since there is less variation in piston speed than there is in revolutions per minute among the various machines, it would seem advisable to use piston speed as the basis of our calculations. Curve *X*, Fig. 193, shows the average of the four piston speed curves, *A*, *B*, *C* and *D*—these being plotted from the present practice of four of the best known builders. By means of the straight lines emanating from these curves, the amount that the present ratings would be effected by making the curve the basis of piston speeds is easily determined. For example point *a* on curve *A* represents 125 tons or A's rating of a certain machine at a piston speed of 342 feet per minute, but since A's displacement rating is less than the suggested 4.4 cubic feet per minute per ton a new rating based on this displacement and the same piston speed would give *a'* or 209 tons and similarly another new rating based on 4.4 cubic feet, and an average piston speed determined from curve *X* as given in table V. would be *a''* or 193.3 tons. The exact effect of this basis of rating on the present ratings of the various machines, from which the curves *A*, *B*, *C* and *D* (Fig. 193) were plotted, is shown in Table VI. From Fig. 193 it will be noted that when the lines connecting the present to the new ratings slant toward the left, it indicates that their ratings have been decreased, and when these lines slant toward the right, that they have been increased by the basis proposed.

32. While it is not claimed that this particular curve, although showing the average practice of four of the best known builders, two being of double and two of single acting machines, should be adopted as the basis from which to figure piston speed, it is intended to show the necessity, and possibly suggest a method of establishing a standard which shall vary with the size of the machines and not distort the ratings as would the use of a constant piston speed or a constant number of revolutions per minute, for machines of all capacities.

33. In establishing a standard of builders' rating, something simple must be agreed upon, so that it may be readily employed by the engineer, the builder and the customer, and above all it should conform as closely as practicable to good average practice.

34. The ideal method of determining the temperatures is by means of thermometers inserted in the liquid and the return lines, which method would eliminate the guesswork connected with liquid temperatures corresponding to head pressures, temperatures of condensing water, etc. With this in view the writer is offering the following basis of nominal rating based on the standard ton of 288,000 British thermal units per day, 12,000 British thermal units per hour and 200 British thermal units per minute, equivalent to the evaporation of 27 pounds of anhydrous ammonia per hour from the standard condition of 90 degrees liquid into 0 degrees saturated gas.

I. *Piston speeds in feet per minute equal or similar to those given in Table V.*

II. *An apparent compressor displacement of* 4.4 *cubic feet per minute per ton, this corresponding to a compressor efficiency of* 90 *per cent. under standard conditions. The equivalent performance of the machine, when working under other conditions, being brought to this basis by the use of factors found in Table II. according to an example already given.*

DISCUSSION.

Prof. F. R. Hutton.—I would call attention to the fact that the Society has in session a committee discussing this question of standard unit of refrigeration for the purpose of presenting a report and recommendation, and that this paper, before being presented at the present meeting, has been referred to that committee and has been approved by them, with the hope that a discussion of the question might throw further light on the subjects considered in the committee.

Mr. Wm. D. Ennis.—In paragraph 11 Mr. Matthews assumes the specific heat of anhydrous liquid ammonia between zero and 90 degrees Fahr. to be 1.2345. What is the authority for this value?* Von Strombeck (*Jour. Frank. Inst.*, Dec., 1890) found a mean value of 1.22876 between 62 degrees and 31 degrees Centigrade. Ludeking and Starr (*Am. Jour. of Science*, III., 45,200) report 0.886. Professor Wood (*Thermodynamics*, seventh ed., page 337) deduces the value 1.093 at a temperature of 34 degrees Fahr. below zero; the same authority gives for the specific

* See foot-note, p. 508.

heat of the liquid at any temperature T degrees Fahr., 1.096—0.0012 T (Trans. A. S. M. E., vol. 10, p. 645) and, at a later date, 1.12136 + 0.000438 T (Trans. A. S. M. E., vol. 12, p. 136). Ledoux gave the formula 1.0058 + 0.003658 t, in which t is the temperature, Centigrade. In 1897, the late Mr. L. A. Elleau and the writer made nine determinations of the specific heat at temperatures between zero and 20 degrees Centigrade (*Jour. Frank. Inst.*, April, 1898). The values obtained ranged from 0.983 to 1.056, the average being 1.02. The probability of error in this or any previous determination is so large that two decimal figures are all that can be asserted with precision. It would appear that the value of the specific heat varies quite widely with the temperature; although so wide a variation as exists between the figures of Von Strombeck and Ludeking and Starr seems improbable. The value used by Mr. Matthews seems to be higher than any that has previously been calculated or determined by experiment. The value of the specific heat of the liquid is not a large factor in calculations in refrigeration; but no correct table of evaporative factors is possible without it, especially as its ratio to the latent heat is about double that of water to its latent heat, and its variation with the temperature is large.

No. 1080.*

CAN A STEAM TURBINE BE STARTED IN AN EMERGENCY QUICKER THAN A RECIPROCATING ENGINE OF THE SAME POWER.†

BY A. S. MANN, SCHENECTADY, N. Y.
(Member of the Society.)

1. If a large steam turbine is cold and at rest how quickly can it be started? Can it be brought up to speed as readily as can a good cross compound engine that is cold all over?

2. Most station men would have doubts as to the adaptability of the large turbine, say 1,500 kilowatts or 2,250 horse-power, for emergency work. So much has been written about the sensitiveness of a rotating disc to the changes of temperature and the effects of unequal expansion that it is easy to imagine difficulties in the rapid start. The possibilities of an engine with a 62-inch low-pressure cylinder in starting practically cold and coming up to synchronous speed are well understood. A station manager would criticise an engineer who would open his throttle as fast as he dared without wrecking his piping system and let his ma-

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*Presented at the Scranton meeting, June, 1905, of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

†For further discussion on this general subject consult *Transactions*, as follows:

No. 345, vol. 10, p. 680: "Notes on Steam Turbine." J. B. Webb.
No. 648, vol. 17, p. 81: "Steam Turbine." W. F. M. Goss.
No. 876, vol. 22, p. 170: "Steam Turbine." R. H. Thurston.
No. 987, vol. 24, p. 999: "Steam Turbine from Operating Standpoint." F. A. Waldron.
No. 1037, vol. 25, p. 716: "Some Theoretical and Practical Considerations in Steam Turbine Work." Francis Hodgkinson.
No. 1038, vol. 25, p. 782: "Different Applications of Steam Turbines." A. Rateau.
No. 1046, vol. 25, p. 1041: "Steam Turbine in Modern Engineering." W. L. R. Emmet.
No. 1047, vol. 25, p. 1057: "De Laval Steam Turbine." E. S. Lea and E. Meden.
No. 1072, vol. 26, p. 389: "Condensers for Steam Turbines." Geo. I. Rockwood.

chine jump into her work. One turn at a time on the throttle is about all that is considered safe, and even then a close watch is kept for groaning valves and cold back bonnets.

3. Every time the starting valve is moved to increase the steam flow the engine is allowed to take its full increment of speed, due to that particular throttle position before the supply valve is moved a second time. There are ten large oil cups, and frequently more, that must be opened and adjusted before the machine moves at all, beside whatever oiling is to be done about the air pumps and other auxiliary apparatus.

4. Most engineers would consider ten minutes as rather a fast start and fifteen minutes as a more usual starting period, including time taken for warming up; in fact, it may not be overstating the case to say that if it were known that an engine driven plant were to be called upon in emergency for power and it were essential that the briefest possible time were to elapse between the call and the taking of the load, one or more engines would be kept in motion all the time, turning slowly and hot all over.

5. This question makes itself very prominent when the steam station is operated as an auxiliary to a large source of high tension power, which is itself in the construction stage and has a large overload capacity of its own to carry, supplying all sorts of apparatus that use electric power, railway, lighting and power circuits simultaneously.

6. At such a time all sorts of accidents will happen to the high-tension water driven plant, most of them due to the necessarily temporary character of many of the electrical connections. It takes months before an intricate system of wiring can be thoroughly relied upon, for it takes months before the temporary work of construction can be replaced.

7. The station at present under consideration is equipped with three Curtis turbine driven alternators, 40 cycle, 10,000 volts, each of 1,500 kilowatts normal capacity. During the summer months the station is operated as an auxiliary to a water power plant, taking all sudden overloads.

8. A signal has been arranged, a $\frac{3}{4}$-inch whistle, so that it can be blown instantly should the power fail. A blast of that whistle means—cut in two turbines and bring the third up to speed. The load will be heavy, and all auxiliary apparatus must be in regular operation.

9. Each turbine has a surface condenser and there are three

or four pumps to be started for each pair of turbines; one circulating pump, one combined hot-well and feed pump, one pressure pump for the step bearings and one dry air pump, all of which are motor driven. The exciter is driven by a steam engine and must be started also, for it supplies current to a portion of the auxiliary apparatus.

10. The boiler room has steam up at all times, supplying a system for manufacturing purposes other than power, and slow fires are kept in enough boilers to make steam needed for the normal load. Forced load means forced fires. The boilers have underfeed stokers, equipped with pressure blast, and will respond quickly to a 50 per cent. excess call for steam. The operating force for this is about equivalent to a force for an engine driven plant. Engineers and oilers, however, are busy about the building on construction work, installing new apparatus and taking such work as their regular occupation when the turbines are not running.

11. At the sound of the whistle the water tender starts a blower on the extra row of boilers: all blast dampers are opened up and all stokers are allowed to feed at the maximum rate. Each fireman dumps his free ash and bars over his red fire.

12. The man in charge of the coal and ash conveyor starts the pressure pump for step bearings. One of the turbine men starts the exciter which supplies current to the auxiliaries beside its field current; a second turbine man starts the circulating pump and then his turbine. The hot-well pump and the air pump are started by the oiler. These movements take place simultaneously. The force is organized upon the lines that obtain in a fire station; each man has his specific duty, and after performing it looks to see that there is nothing more for him to do. Only a few seconds elapse between starting the first pump and starting the first turbine.

13. The turbine throttle is opened as fast as an 8-inch steam valve can be opened without endangering the steam piping system. It is not considered advisable to open the throttle valve as fast as a man's strength will permit; but if nothing unusual occurs in the pipe line, sentiment does not spare the turbine.

14. One electrician attends to the switchboard and telephone. As soon as the machine approaches speed, the synchronizing system is cut in and the main switches are got ready. One and one-half minutes will do all the work here outlined, including the time

taken in mustering the crew from various parts of the building, itself not a trivial matter.

15. Manipulating an engine regulator so that it shall be at a precise speed and at an exact phase relationship from some other machine, not more than 1-1500 part of a second removed from it, is no matter that can be hurried, and one minute is fast time on such work. But the whole thing, phasing-in and all, has been done in 2½ minutes, including full load on the turbine, which started from a standstill.

16. This performance has been gone through a great many times, and our record book shows that out of 43 such calls, 10 starts were made in 2½ minutes, 18 in 3 minutes and 15 in 3½ minutes.

17. We have taken the time in a number of instances when all the auxiliaries have been in motion and it only remained to start the turbine and phase it in on the line: the only valves to open in such cases are the throttle and one small oil valve. The two quickest starts have been made in forty-five seconds and seventy seconds, respectively, including phasing in. Others range between one minute ten seconds and one and one-half minutes. These two quickest starts were made on a turbine which had stood for twenty-four hours with the throttle valve shut tight though there was a slight leakage past the seat. After the throttle valve is off its seat it is not more than thirty seconds before the turbine is up to speed. A cross compound reciprocating engine of the four valve type, 2,250 horse-power capacity, can be brought up to speed from a standstill in five minutes if it is hot all over. This five minutes is to be compared with the seventy seconds required for the similar turbine operation.

18. A reciprocating engine, which is turning over slowly with the throttle valve just off its seat or with by-pass open and having all its oil cups open and regulated, can be brought up to speed, say seventy-five turns, in two and one-half minutes. This can be compared with the thirty seconds necessary for bringing the turbine up under the same conditions; that is, about one-fifth the time necessary for bringing up the engine.

19. If the engine is cold all over and has all its oil cups shut tight, all its auxiliaries quiet, fifteen minutes is called a rapid start. Starts have been made under such conditions in twelve minutes. When we start a cold turbine, we open up the valve and let her turn, and in two minutes we are ready to bring her up

to speed and she will be at speed in two and one-half minutes, dividing the engine's time by more than four.

DISCUSSION.

Mr. E. H. Foster.—Has Mr. Mann had experience with starting reciprocating engines with steam superheated about 100° F.? I have gathered that one of the unexpected benefits to be derived from superheated steam is the ability to start up engines quickly. This result has been observed in Corliss as well as slide valve engines.

Mr. Richard H. Rice.—This paper gives an evidence of the apparently rough treatment to which the turbine can be subjected without doing it any damage. At first sight those who have been accustomed to handling other classes of machines would say that this was treatment which ought not to be given any apparatus. As a matter of fact, there is no reason why turbines should not be handled indefinitely in this way. It may be interesting to state that it is the custom in the works with which I am connected, when starting up a 1500 kw. turbine for the first time, to bring it up to speed in fifteen minutes from the time steam is first turned on. It is possible to do this in view of the fact that the shaft revolves in a vertical position and the pressure on the bearings is therefore negligible.

*Mr. A. S. Mann.**—Since writing this paper the 1500 kw. turbines described have been started and phased in on the line in thirty seconds after having stood twelve hours. The writer knows of no equal performance on reciprocating engines with either saturated or superheated steam.

*Author's Closure, under the Rules.

No. 1081.*

COUNTERWEIGHTS FOR LARGE ENGINES.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

1. The writer was called upon to investigate the problem of lessening the vibration of the engines at the new power plant at the Manchester St. station of the Rhode Island Co., Providence, R. I., and working conjointly with Mr. Fred N. Bushnell, the chief engineer of the company, he recommended that certain counterweights be added in addition to those already on the engines. These counterweights which were exceptionally large were adopted and gave satisfactory results.

2. In addition to the problem of lessening the vibrations certain electrical problems arose regarding the effect of the counterweights in increasing the variation in angular velocity of alternating current generators run in parallel with each other. A number of measurements were made to determine the action of the counterweights, both as affecting the vibration of the building and the tendency of the alternating current generators to break apart when run in parallel, and as these data may be of interest to the Society they are hereby presented.

3. The counterweights were applied to three engines, all of which were of the same cylinder dimensions and had the same weights of reciprocating parts. The engines were built by the Filer and Stowell Co. of Milwaukee, Wis., and were of the horizontal cross compound type directly connected to electrical generators. Two of the engines, designated as Nos. 1 and 2, were connected to 1,500 kilowatts alternating current generators, and one, designated as No. 3, to a 1,600 kilowatts direct current generator. A photograph of a portion of the revolving field of the

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

No. 2 engine showing one of the counterweights which was placed in the revolving fields is reproduced in Fig. 194.

4. On inspecting the plant it was evident that something should be done to diminish the shaking. The entire building rested on piles which passed through silt for about 30 feet before striking a solid bottom. There was a mass of concrete about 5 feet thick placed on these piles which extended under the entire plant so as to form a bed on which rested the foundations of the buildings,

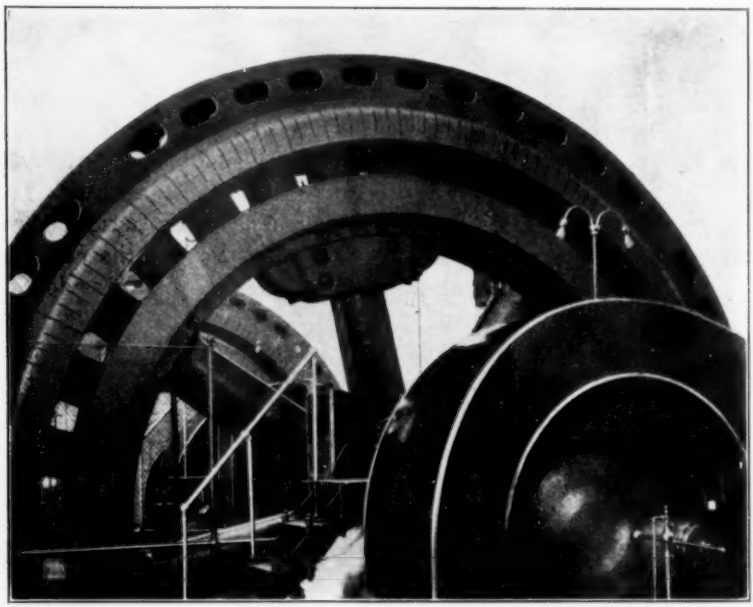

FIG. 194.—REVOLVING FIELD OF NO. 2 ENGINE SHOWING COUNTERWEIGHT.

boilers and engines. The bed of concrete also extended under the chimney foundation. When the first observations were made the buildings were not completed. All three engines were, however, installed and were protected by a temporary wooden structure. The boiler plant was in the process of construction, much of the brick work having been laid, and the steel framework of the building nearly completed. A portion of the boiler plant was running to supply steam to one of the engines. The chimney, which was ultimately to be about 300 feet high, was being erected. This was built of special brick furnished by the Alphons Custodis Chimney Construction Company.

5. The engine which was running was that designated as No. 3, and it shook the entire foundation to such an extent that the motion could be readily felt by a person standing thereon. When water was poured outside of sheet piling, which surrounded the foundation, the surface of the water could be seen to move up and down in time with the strokes of the engine.

6. After inspecting the plant the writer constructed an apparatus on the principle of the seismograph for measuring the amount of vibration. On a second visit to the plant the horizontal shake of the foundation was measured with this instrument with engine No. 3 running, and found to be 0.01 inch at the engine. This may not seem to be a large amount, but as the entire mass of the building foundation shook, the movement of parts of the temporary wooden building was magnified in many places so as to be very evident to the eye. Furthermore the iron work of the boiler house shook considerably. That the entire mass of the foundation shook could be appreciated by feeling the vibration through the feet and also by measuring the actual amount at different points with the special apparatus. At a point near the extreme end of the foundation where the No. 1 engine was located the foundation was found to shake 0.008 inch. Measurements made near the top of the chimney, which was erected to the height of about 175 feet above the ground, showed that the maximum shake with the engine running at its ordinary speed of 90 revolutions per minute was about 0.02 inch. After measuring the vibration of the chimney with the engine running at its ordinary speed the engine was shut down and a marked result took place when its speed fell in harmony with the time of vibration of the chimney. When this occurred, the chimney shook to such an extent that the motion was beyond the range of the special instrument. The total movement of the pointer of the instrument was such that the chimney was shown to move more than $\frac{1}{8}$ of an inch. In constructing the chimney the workmen had noticed that when they came to a height of about 130 feet the vibration was much greater than it was after the chimney was built higher. This made it appear that at the height of 130 feet, at which there was the most shake, the time of the vibration of the chimney was in harmony with the number of revolutions made by the engine.

7. As the entire plant shook laterally on top of the piles, and as there could be no vertical movement, it appeared evident that the best plan would be to place counterweights on the engines of

such sizes that they would diminish the horizontal shaking forces to nearly a minimum irrespective of the vertical shaking forces which might be produced. This would give much larger counterweights than are ordinarily used, but on carefully considering the conditions which existed it was deemed best to adopt them. The counterweights originally on the engines were comparatively small and it was found impossible to place large enough weights on the crank disks to carry out this plan. It was therefore decided to

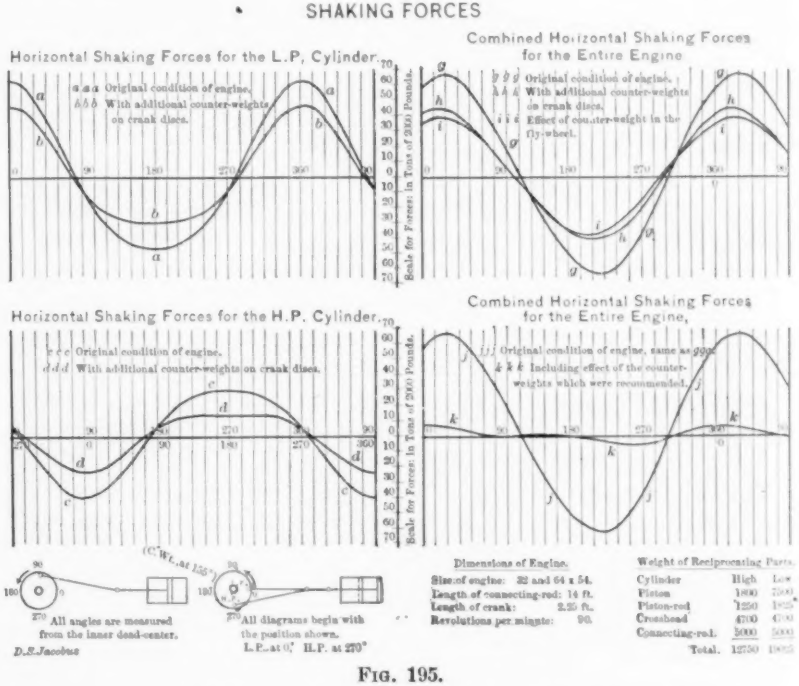
SHAKING FORCES
Horizontal Shaking Forces for the L.P. Cylinder.
a a a Original condition of engine.
b b b With additional counter-weights on crank discs.
Combined Horizontal Shaking Forces for the Entire Engine
g g g Original condition of engine.
h h h With additional counter-weights on crank discs.
i i i Effect of counter-weight in the fly-wheel.
Scale for Forces: in Tons of 2000 Pounds.
Horizontal Shaking Forces for the H.P. Cylinder.
c c c Original condition of engine.
d d d With additional counter-weights on crank discs.
Combined Horizontal Shaking Forces for the Entire Engine.
j j j Original condition of engine, same as ggg
k k k Including effect of the counter-weights which were recommended.
All angles are measured from the inner dead-center.
All diagrams begin with the position shown. L.P. at 0° H.P. at 270°
D.S. Jacobus
Dimensions of Engine.
Size of engine: 32 and 64 x 54.
Length of connecting-rod: 14 ft.
Length of crank: 2.25 ft.
Revolutions per minute: 90.
Weight of Reciprocating Parts.
Cylinder High 1800 Low 7000
Piston 1800 7000
Piston-rod 1250 1875
Crosshead 4700 4700
Connecting-rod 3000 5000
Total. 12750 19875

FIG. 195.

place counterweights on the fly wheels in addition to placing as heavy counterweights as possible on the crank disks. A counterweight in the fly wheel acts just as efficiently as one placed on the crank disk in eliminating the shaking forces tending to translate the bed of the engine, but with such a counterweight there remain forces which act as couples and tend to shake the engine by rotating it about its center of mass. The horizontal forces tending to produce translation were, however, the most important in the case under consideration, and these were considered irrespective of the forces tending to produce rotation.

8. The weight of the counterweights recommended to be added in addition to those already on the No. 3 engine was 4,995 pounds on each of the crank disks and 3,600 pounds in the fly wheel, the distance of the center of gravity of the weights on the crank disks from the center of the shaft being 2.38 feet and for the weight in the fly wheel 7.67 feet. The counterweights originally in each crank disk amounted to a net weight of 1,740 pounds at 2.05 feet.

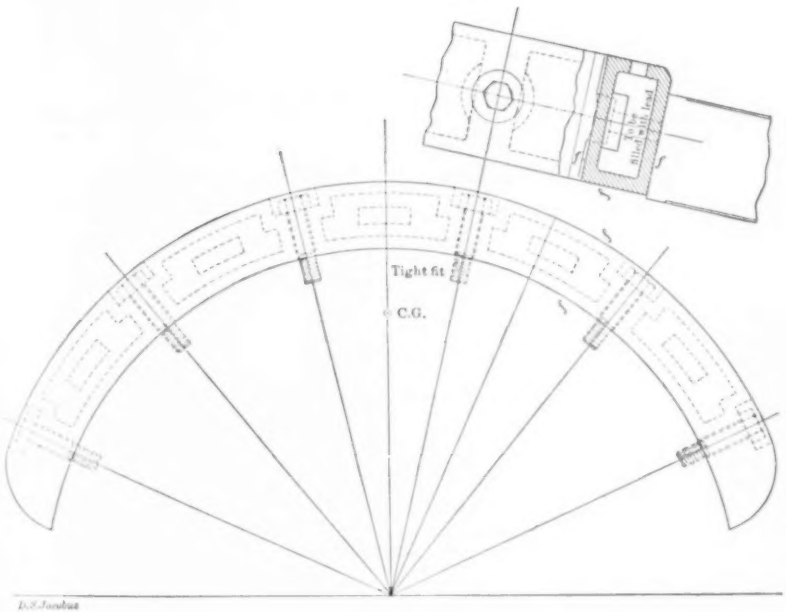

FIG. 196.

9. The position of the fly wheel counterweight and the shaking forces with and without the additional counterweights are shown in Fig. 195.

10. Mr. Bushnell designed counterweights for the No. 3 engine to conform with the above sizes. These counterweights are shown in Figs. 196 and 197. The counterweights finally placed on the engine were somewhat lighter than called for but not enough so to produce any great difference in the results.

11. On starting up the No. 3 engine with the counterweights it ran without appreciable vibration and operated with entire satisfaction.

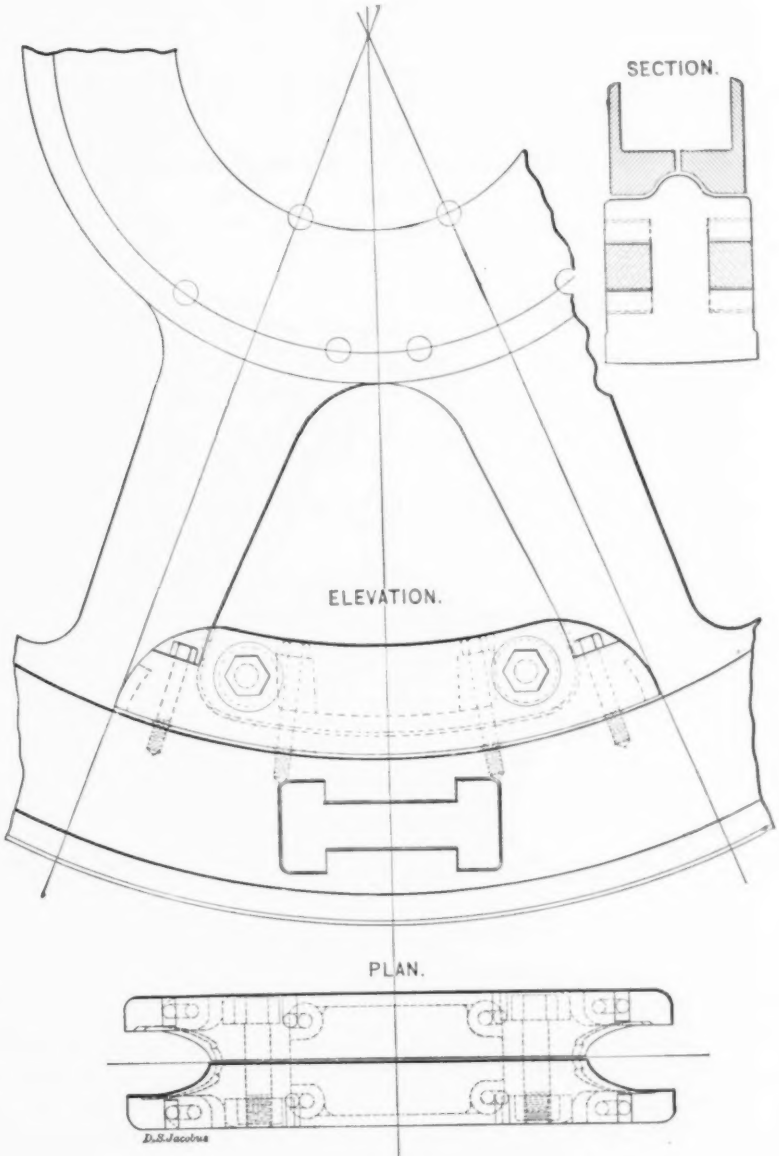

FIG. 197.

12. Counterweights were then placed on engines Nos. 1 and 2 driving the alternating current generators. As the dimensions of these engines and the weights of the reciprocating parts were

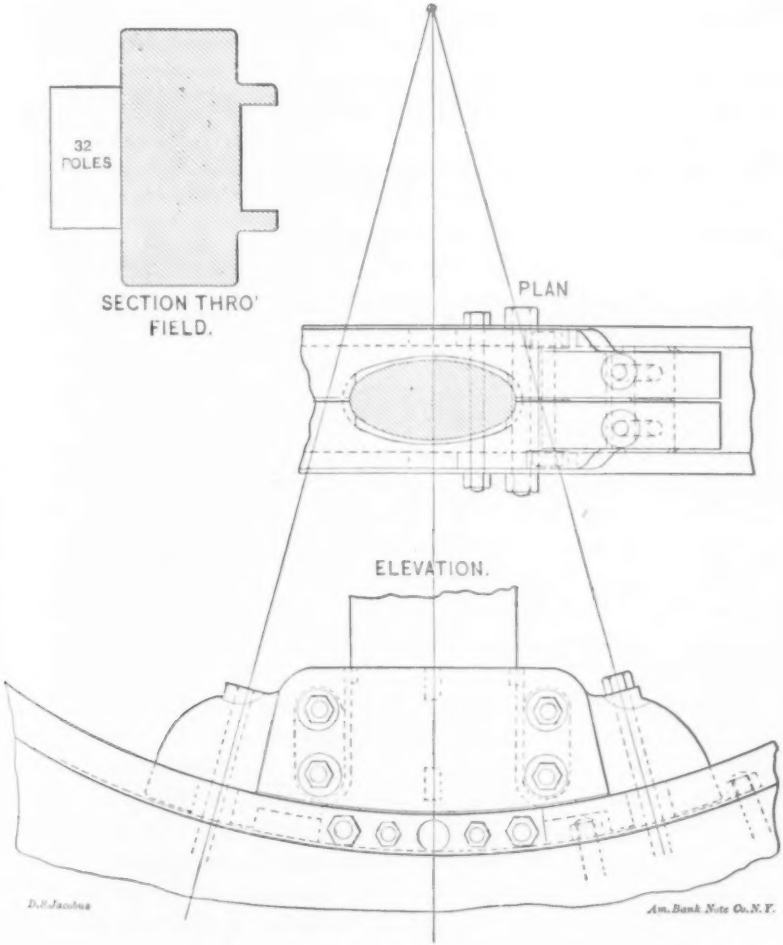

FIG. 198.

the same as for the No. 3 engine, the same size of counterweights were used as for the latter, but those placed in the revolving fields were necessarily of a different form than that employed in the fly wheel of the No. 3 engine. In these engines the high pressure crank leads; whereas, the low pressure leads in the No. 3 engine,

and this was taken account of in placing the counterweights in the revolving fields. The counterweights for the revolving fields of engines Nos. 1 and 2 are shown in Fig. 198.

13. There was some delay in arranging for a third visit to the plant. The buildings and the chimney of the plant were then completed. A vertical engine directly connected to a 2,500 kilowatt generator had been erected and was running in addition to the three horizontal engines. It was found that the greatest shaking that could be noticed occurred on the switchboard platform which was anchored to one of the side walls of the building at a point some distance above the floor. When engines Nos. 1 and 2 driving the 1,500 kilowatts alternators were run in parallel with the counterweights opposed they produced but little vibration. When run with the counterweights together there was a greater vibration. The greatest vibration that could be detected on the switchboard platform with the counterweights of the two alternating machines together and all the engines in the station running, including Nos. 1, 2 and 3, and a vertical engine which is designated as No. 4, was about 0.023 inch. This maximum shake occurred when the counterweights of the three engines Nos. 1, 2 and 3 fell together, which would take place about once every 15 seconds, as the two engines running the alternators ran at about 94 revolutions per minute and that driving the direct current machine at 90 revolutions per minute. The amount that the floor shook was measured and was found to be extremely small and less than 0.002 inch.

14. The ground was frozen during the above test and it was decided that it would be best to make a final set of observations of the amount of shaking after the frost was entirely out of the ground. Furthermore, some questions arose as to the possible danger of the alternating current generators breaking out of step when run in parallel with their field counterweights opposed, because when so run, the ammeters attached to the generators swayed considerably, indicating fluctuations in the current.

15. On looking through the magnets of the revolving fields of one of the alternating current generators at the revolving field of the other the angular variation back and forth was apparent. This was carefully estimated and under the ordinary conditions of running was found to amount to from about 6 to 8 pole degrees total variation or from about 3 to 4 pole degrees for each machine. The amount of this angular variation with the machines running

in parallel was about the same irrespective of the position of the counterweights with reference to each other, but when the counterweights were opposed the variations occurred every stroke or about 94 times per minute; whereas, when the counterweights were together, or nearly so, the variations occurred at less frequent intervals, the total number of such variations then being from about 30 to 40 per minute. As the total displacement of the two fields from their true position, as observed by looking through the magnets of one revolving field at the other, was about the same irrespective of the relative positions of the two counterweights, it follows that this displacement was produced as much through governor action as through any variation in speed during a single stroke.

16. When the two alternating machines were operating in parallel there was a swaying of the ammeters in the machine circuits. This swaying was, however, of an entirely different frequency from that of the strokes of the engines. The amount of this swaying was noted for different relative positions of the counterweights of the engines. The engines were operating at the time under light loads. The voltmeter reading was also noted and found to be steady in every instance.

17. It appeared that the fluctuations in the current generated by the two alternating machines had but little influence on their efficiencies and that they were working successfully when connected in parallel. The writer did not wish, however, to give an opinion on the electrical side of the problem and arranged with Prof. Albert F. Ganz, the Professor of Electrical Engineering of the Stevens Institute, to accompany him at the time of making the final measurements.

18. On a joint visit of Professor Ganz and the writer observations of the angular displacement of the rotating fields and electrical readings were again taken and these observations led Professor Ganz to conclude that there was no apparent danger of the alternators breaking out of step while operating in parallel with the counterweights in any relative position.

19. Careful measurements were also made at this time of the amount that the engines shook the building. Observations of the angular displacement of the revolving fields when run in parallel bore out in a general way those made on my previous visit.

20. In order to study more thoroughly the effect of the counterweights in increasing the angular variation, diagrams of crank

efforts were laid off corresponding to indicator cards taken from the engines and from these curves of crank effort the angular variation in pole degrees was determined, as shown in Fig. 199. From this figure it may be seen that the variation for a single machine without the addition of the counterweights in the revolving fields

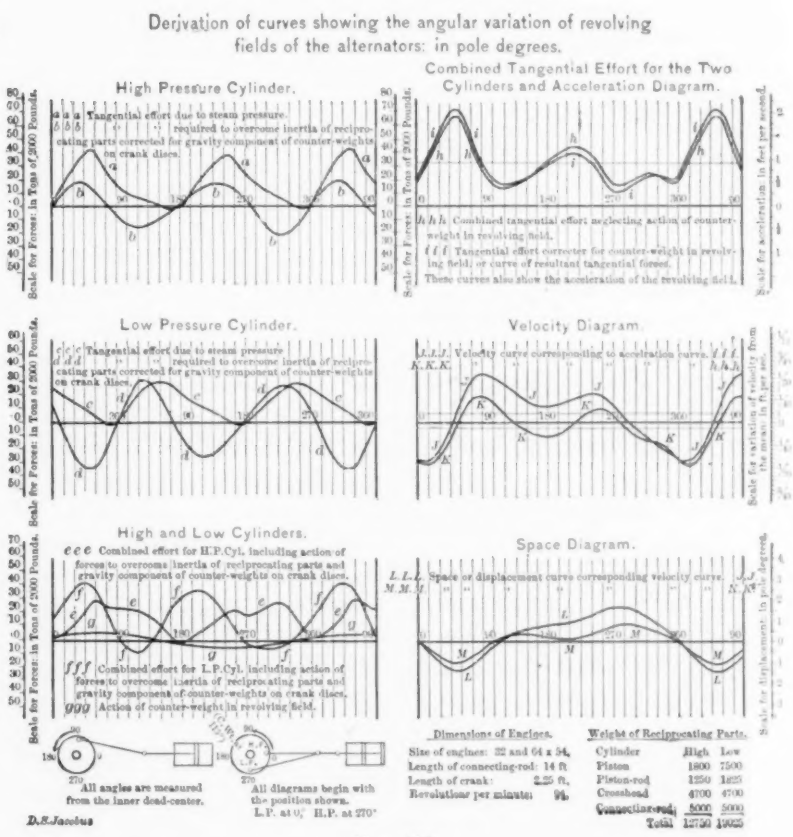
Derivation of curves showing the angular variation of revolving fields of the alternators: in pole degrees.
High Pressure Cylinder.
Combined Tangential Effort for the Two Cylinders and Acceleration Diagram.
Low Pressure Cylinder.
Velocity Diagram.
High and Low Cylinders.
Space Diagram.
Scale for Forces: in Tons of 2000 Pounds.
Scale for acceleration: in feet per second.
Scale for variation of velocity from the mean: in ft. per sec.
Scale for displacement: in pole degrees.
All angles are measured from the inner dead-center.
All diagrams begin with the position shown. L.P. at 0°, H.P. at 270°
D.S.Jacobus
Dimensions of Engines.
Size of engines: 32 and 64 x 54.
Length of connecting-rod: 14 ft.
Length of crank: 2.25 ft.
Revolutions per minute: 94.
Weight of Reciprocating Parts.
Cylinder High 1800 Low 7500
Piston 1250 1625
Piston-rod 4700 4700
Crosshead 5000 5000
Connecting-rod
Total 12750 19825

FIG. 199.

would be 1.8 pole degrees, and with the counterweights 3.2 pole degrees. The latter value is the one to be compared with the variations observed by looking through one of the revolving fields and observing the other, and it agrees with the amount which was observed, which varied from 3 to 4 pole degrees.

21. It appears from Fig. 199 that by placing the counterweights

in the revolving fields the angular variation during a single stroke was made about twice what it would have been without the counterweights. It however appears that the increased angular variation during a single stroke due to employing the counterweights does not introduce any great disadvantage as the alternators can be safely run in parallel at practically the same efficiency as would exist should there be less angular variation.

22. As already stated the counterweights actually placed in the engines were of a less weight than the amounts called for but not enough so as to produce any great difference in the results obtained. They were also placed at different radii from those called for. The weights of the counterweights and their distances from the center of the shaft which were recommended and which were actually used as follows: For comparing these the product of the weights into the distances of their centers of gravity from the center of the crank shaft are given as these products are a measure of the effects produced on the shaking forces.

	Recommended.	Actually used.
Engines Nos. 1 *and* 2 *driving alternating current generators:*		
Counterweights placed on crank disks	4,995	4,100 lb.
Counterweights placed in revolving fields	3,600	3,467 "
Distance of center of gravity of the weight in the revolving fields to the center of the shaft	7.67	7.56 ft.
Distance of center of gravity of the weights on the crank disks to the center of the shaft	2.38	2.77 "
Product of weight in tons by the distance in ft. — Crank disks counterweights	5.94	5.68 "
Product of weight in tons by the distance in ft. — Revolving field counterweights	13.81	13.11 "
Engine No. 3 *driving direct current generator:*		
Counterweight placed on crank disk of high-pressure cylinder	4,995	4,100 lb.
Counterweight placed on crank disk of low-pressure cylinder	4,995	4,180 "
Counterweight placed in flywheel	3,600	3,600 "
Distance of center of gravity of weight in the flywheel to the center of the shaft	7.67	7.06 "
Distance of center of gravity of the weights on the crank disks to the center of the shaft	2.38	2.77 "
Product of weight in tons by the distance in ft. — Crank disk Counterweights (high-pres. cyl.)	5.94	5.68
Product of weight in tons by the distance in ft. — Crank disk Counterweights (low-pres. cylinder)	5.94	5.79
Product of weight in tons by the distance in ft. — Flywheel counterweights	13.80	12.71
Weight of original counterweights on crank disks of all the engines	....	1,740 lb.
Distance of center of gravity of the original counterweights from the center of the shaft	....	2.05 ft.

VIBRATION OF FOUNDATIONS AND OF SWITCHBOARD PLATFORM.

The height of the diagram measured from one tooth or wave to the bottom of the one next following represents the amount of vibration.

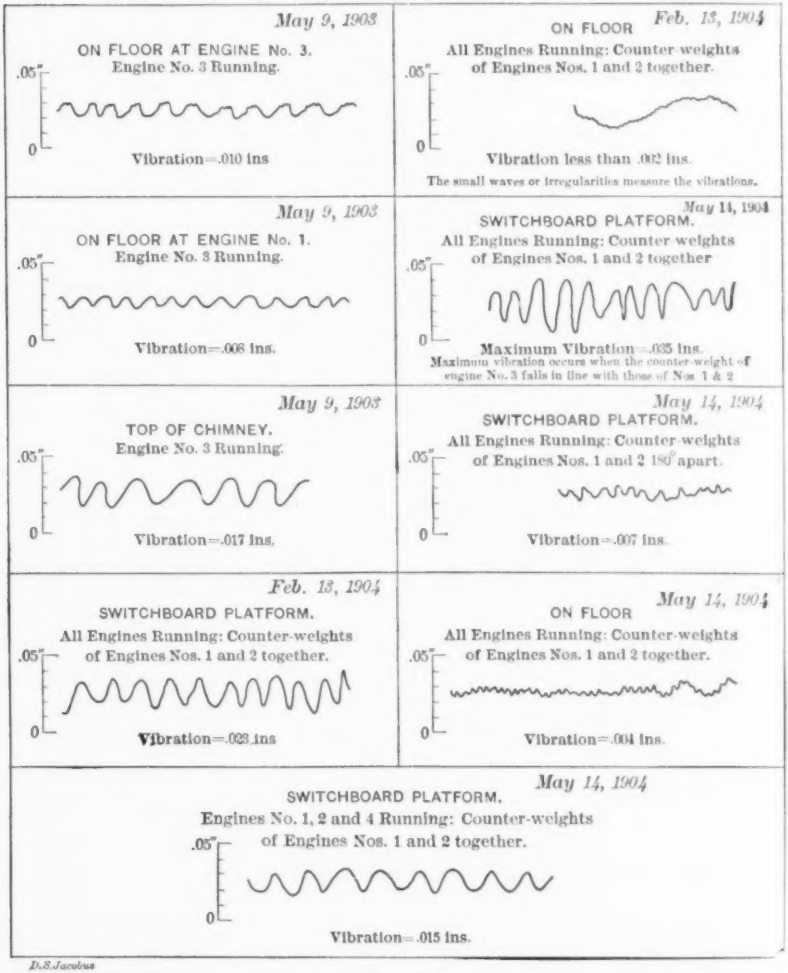

FIG. 200.

23. Fig. 200 shows characteristic records taken with the special instrument for measuring the amount of shake. The principle on which this instrument was constructed is shown in Fig. 201. A light pointer A was made to rest against a large mass suspended from a point at some distance above. In most of the experiments this mass was the lower block of the hoist of a traveling crane.

This large mass was not affected by the vibrations of the building, or if affected its vibrations had a different period from those to be measured. The instrument was mounted at a part of the building where the vibration was to be measured and therefore moved in the same way as the building. In taking a record the drum *B* was revolved by hand by pulling the cord *C* and a curve was traced by a pen *D* on a paper placed on the drum *B*. The height of the diagram traced by the pen represented the amount of vibration. In the diagrams the vibrations were magnified fifteen times. The fact that the cylinder was rotated by hand accounts for some

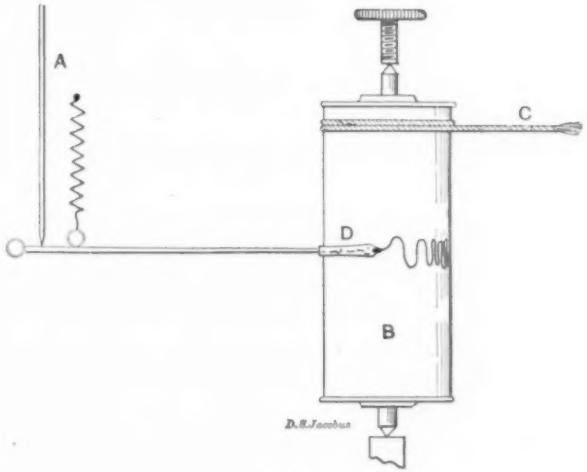

FIG. 201.

of the irregularities in the curves, but this does not affect the accuracy of the records which depend only on the height of the waves measured from the top of one to the bottom of the one next following.

24. It may be seen from Fig. 200 that the amount of vibration was greater at the time of the last visit than it was when measured on February 13th. This is accounted for by the fact that the ground about the building was frozen on February 13th, whereas such was not the case when the final measurements were made. The final measurements gave 0.035 inch for the maximum vibration of the switchboard platform with all the engines running and with the counterweights of engines Nos. 1 and 2 together. The movement of the floor of the engine room and of

the foundation for this set of conditions was 0.004 inch. The amount that the foundation shook with the No. 3 engine running alone without counterweights, as measured on the first visit, was 0.01 inch. It therefore follows that there was less than one-half the amount of vibration with all three engines running with their counterweights in such a position that they gave the maximum shake than there was with a single engine running without the additional counterweights.

25. After the above investigation was completed Mr. Bushnell made observations with a transit to determine whether there was any shaking of the chimney with all the engines running and found that there was no perceptible motion.

26. The method of computing the size of the counterweights and the angular displacement of the revolving fields of the alternating current generators when run in parallel is given in the appendix which follows.

APPENDIX.

METHOD OF COMPUTING THE SIZE OF COUNTERWEIGHTS REQUIRED.

27. To most nearly balance the horizontal shaking forces in one of the engines a counterweight equivalent to the entire mass of the reciprocating parts with its center of gravity at the crank radius should be used. It was impossible to place such a mass on the crank disk, and, as has already been stated, as heavy counterweights as possible were placed on the crank disks and an additional counterweight was placed either in the fly wheel or in the revolving field.

28. The maximum counterweights which could be placed on the crank disks were estimated at 4,995 pounds at 2.38 feet. The counterweights originally on the engines amounted to 1,740 pounds at 2.05 feet, so that the counterweights originally on the crank disks, together with the counterweights added to the crank disks, amounted to 6,870 pounds at the crank radius. As the total weight of the reciprocating parts amounted to 12,750 pounds for the high pressure cylinder and 19,025 pounds for the low pressure cylinder there remained 5,880 pounds at the crank radius to be provided for by counterweight action in the high pressure cylinder and 12,155 pounds at the low pressure cylinder. To accomplish this action a counterweight should be placed in the fly wheel of

3,960 pounds at 7.67 feet from the shaft center. For the direct current generator where the low pressure crank leads this counterweight should be placed about 155 degrees ahead of the low pressure crank, and for the alternating current generators where the high pressure crank leads it should be placed about 115 degrees ahead of the high pressure crank.

29. The full counterweight was not, however, recommended, as a substantial balance of the horizontal forces would be obtained by employing a weight of 3,960 pounds in the fly wheels or revolving fields instead of 4,000 pounds, and it was considered advisable to keep slightly within the theoretical limit.

30. In plotting the shaking forces given in Fig. 195 the gravity component of the rod at the crank pin end is assumed to revolve with the crank pin and the remainder to have the same motion as the piston.

31. As the distance of the center of gravity of the rod from the crank pin was 46 per cent. of the length of the rod this gave 2,700 pounds for the revolving part at the crank pin and 2,300 pounds for the part which was assumed to move with the piston. This gives correct results as far as the shaking forces are concerned. In obtaining the horizontal shaking forces for the counterweights the centrifugal force is multiplied by cos θ where θ is the crank angle measured from the line of centers. For the mass moving along with the piston the horizontal shaking forces were obtained by the formula*

$$F_2 = m r^2 R \ (\cos \theta + Z),$$

where Z can be given the approximate value of

$$\frac{\cos 2\theta}{n}.$$

METHOD OF DETERMINING THE ANGULAR DISPLACEMENT OF THE REVOLVING FIELDS OF THE ALTERNATING CURRENT GENERATORS WHEN RUN IN PARALLEL.

32. This work is represented by the curves shown in Fig. 199. The tangential effort curves due to the steam pressure which are there given were obtained from indicator cards taken from the engines. The effect of the inertia of the connecting rod on the

* For notation and formula see article by the author, *Transactions* of the American Society Mechanical Engineers, Vol. XI., pages 513 and 515.

tangential effort is obtained by assuming that one-half of the mass of the rod is concentrated at the crank pin and one-half at the crosshead.* The inertia forces are worked out by the formula already given and the tangential effort is worked out in the ordinary way. The effect of the gravity components of the counterweights was included in the analysis. Two sets of results were worked out, one for the engines as they are now running with the counterweights which were added, and one assuming that the counterweights in the revolving fields were removed.

33. The final curve of tangential effort for the entire engine if measured on the proper scale represents the acceleration. This scale was determined as follows:

34. The fly wheel action of the revolving parts amounted to 2,236,000 pounds at the crank radius.

35. The scale of the ordinates of the diagrams as originally constructed was 64,000 pounds per inch and the ordinates were ¼-inch apart, each space representing 15 degrees of crank angle.

36. The acceleration corresponding to one inch height of the ordinates was therefore

$$64{,}000 \times 32.2 \div 2{,}236{,}000 = 0.922 \text{ feet per second},$$

which gave the scale laid off to the right of the acceleration diagram.

37. The areas included between each two consecutive ordinates of the acceleration diagram were measured in square inches and this area gave the increments in velocity for the velocity diagram.

38. In laying off the increments in the velocity diagram the number of square inches were multiplied by two and laid off in lineal inches. If f represents the acceleration, $\triangle t$ the time corresponding to the distance between two of the ordinates, or for the crank to turn 15 degrees, and $\triangle v$ the variation in the velocity, we have

$$f \triangle t = \triangle v.$$

As the crank revolved at 94 turns per minute, we have

$$\triangle t = 60 \div (94 \times 24) = 0.0266 \text{ sec.}$$

As the space between the ordinates of the acceleration curve was ¼ inch and the scale of the ordinates 0.922 feet per sec., the

* This is the way recommended by Mr. Sanford A. Moss in a paper presented at the last meeting of the Society. A note on Mr. Moss' paper follows.

value of f would be obtained by multiplying the area between two of the ordinates by 0.922 x 4 or by 3.688, hence we have

$$\triangle v = \text{Area between two ordinates} \times 0.0266 \times 3.688 = 0.0981.$$

39. The area between the ordinates was multiplied by two in laying off the increments in the velocity diagram, hence for one inch of height in the velocity curve would be $0.0981 \div 2 = 0.0491$ feet per second.

40. The areas measured in square inches on the velocity diagram were laid off directly in lineal inches in the space diagram. The scale for this diagram was therefore

$$4 \times 0.0491 \times 0.0266 = 0.00522 \text{ feet per inch of height.}$$

41. To obtain the scale in pole degrees the circumference of the crank circle or 14.14 feet was divided by 16, the number of poles and the distance measured between the poles at the crank circle was found to be 0.884 feet. The distance per pole degree was $0.884 \div 360 = 0.00246$ feet. The scale of the diagram in pole degrees per inch was therefore

$$0.00522 \div 0.00246 = 2.12.$$

NOTE ON PAPER PRESENTED AT THE LAST MEETING OF THE SOCIETY BY MR. SANFORD A. MOSS ON THE INFLUENCE OF THE CONNECTING ROD ON THE ENGINE FORCES.

42. The writer's first impression on looking over this paper was that it could contain no new matter as the subject has been gone over many times before. For example, the papers presented to the Society of Naval Architects and Marine Engineers by Admiral George W. Melville and by Naval Constructor D. W. Taylor, and forming part of Vol. IX., 1901, of the Transactions of that Society are masterpieces of literature on this subject. Again, in a paper presented by the author at the New York meeting in 1889, and which was published in Vol. XI. of the Transactions, the exact forces required to accelerate a connecting rod are derived and numerical values worked out for several classes of engines. In this paper the effect of friction was also computed, that for the connecting rod being obtained by a method devised by Prof. J. Burkitt Webb and the writer and published in the Annals of Mathematics for December, 1888.

43. On examining Mr. Moss' paper carefully, however, it appeared that he has demonstrated that the mass of the connecting rod may, without undue error, be divided between the wrist and crank pins so that by considering only that part which he assumes to be placed at the wrist pin the tangential effort at the crank pin may be accurately determined. In other words it follows from Mr. Moss' work that with connecting rods of ordinary construction, if one-half of the mass of the rod is assumed to move with the piston and cross head, and no other allowance be made for the forces required to accelerate the rod, the values obtained for the tangential effort exerted on the crank shaft will be substantially correct. This certainly gives an easy way of handling the forces for the rod in determining the tangential forces at the crank pin and its advantage over more complicated methods cannot be disputed. This method has been employed in the present paper.

44. Mr. Moss' analysis does not, however, furnish any new light in the selection of proper counterweights for an engine and, indeed, his deductions regarding a counterweight are apt to be misleading. He discusses counterweights and shaking forces and does not point specially to the tangential force at the crank pin, or the force transmitting the power, which involves the one feature in which his analysis excels others.

45. In Mr. Moss' analysis he determines what he calls the average centrifugal force of the rod and states that if we counterbalance this we eliminate its tendency to shake the engine bed. It is true that we eliminate the tendency of the so called centrifugal force to shake the bed, but this is only one of several forces exerted on the bed, and in estimating the size of a counterweight all the forces which act on the bed should be considered. The only forces in a connecting rod which need ordinarily be considered in estimating the size of a proper counterweight are those which produce translation of the center of gravity as the forces which govern the angular movement necessarily occur in couples which cannot be balanced by a single counterweight. When the translative forces are considered the vertical components in a horizontal engine will be balanced by a counterweight equivalent to the gravity component of the connecting rod at the crank pin. Mr. Moss states that if a counterweight to balance what he calls the radial or centrifugal force is provided, the engine will be exactly in equilibrium in the direction at right angles to its axis. This is not so, for in order to obtain such an equilibrium the

translative forces at the center of gravity of the rod should alone be considered and the proper counterweight would be one which would balance the gravity component of the rod at the crank pin, whereas the counterweight specified by Mr. Moss for rods of ordinary construction would, according to his own estimate, be about eight-ninths of this.

DISCUSSION.

Mr. Fred. N. Bushnell.—The methods described in this paper for lessening the vibration of the engines are of great interest and value to engineers who have to do with the installation of machinery where the nature of the soil is such that is is impossible to get absolutely rigid foundations within commercial limits of cost. The conditions in Providence are such as exist in many other cities where large areas of land along the water fronts have been made by filling in with gravel, cinders and miscellaneous refuse, and a more extended description of the underlying earth strata and the character of the sub-structure in this case may be of interest.

The Manchester Street power station is situated upon the westerly side of the Providence river, with its easterly end about fifty feet from the harbor line. Originally this site was covered with water, but some years ago it was filled in to about elevation 5 above mean high water. Before preparing the plans for the foundations, a number of borings were made within the area to be occupied, and the nature and average depth of the earth strata were ascertained to be as follows:

Sand, gravel and cinders (filling), 13 feet;
Silt, 28 feet;
Sand, gravel and a little clay, 4 feet;
Very fine sand and a little clay, 25 feet.

Underlying all of the above was a deep bed of clay, sand and gravel.

The conditions revealed by these borings indicated that it would not be advisable to carry the foundations down to the solid substratum on account of the cost, and it was therefore decided to build a pile and concrete sub-structure of ample strength for the loads to be imposed, and to provide as far as possible against vibration by enclosing it within a heavy sheet-piling cofferdam which would present a large bearing surface to the surrounding earth.

While it was hoped that this form of construction would prevent objectionable movement of the building, it was realized that the resisting power of the earth could not be determined with absolute certainty, and in deciding upon the plans, the possibility of having to take some measures later on to lessen the horizontal shaking forces in the engines received proper consideration. The sheet-piling cofferdam was made of 6-inch Georgia pine, driven to a depth of forty feet below mean high water. It enclosed the entire area of the building, approximately 33,500 square feet. This cofferdam, in addition to providing against vibration of the building, was also intended to exclude all water from the enclosure during the progress of the work. Within this area 5,475 spruce and chestnut piles were driven, spaced thirty inches from center to center, excepting under the steel columns and the chimney, where they were spaced on 24-inch centers. The longest of these piles was driven to a depth of sixty-three feet below mean high water, and their average length was fifty-one feet. No pile was allowed to be driven which was less than six inches in diameter at the tip. The pile cut-off under the chimney was at elevation –7, and under the rest of the building at elevation –2. The earth was excavated to a depth of about one foot below the pile cut-off, and a bed of Portland cement concrete was laid around and on top of the piles to elevation 2, covering the entire area enclosed by the sheet-piling cofferdam. This work constitutes the substructure supporting the building walls, chimney, foundations for engines, boilers, etc. The pressure of the earth on the outside surfaces of the sheet-piling cofferdam, however, did not prove to be quite sufficient to hold the structure against the shaking forces in the engines, and it was decided to investigate the nature of these forces and to apply such remedies as were necessary. Professor Jacobus' paper fully describes the ingenious instrument which he devised to accurately measure the vibrations, and also the methods employed to determine the proper counterweights to apply to the disc cranks and fly wheels to correct it. In practice the results obtained from the application of these counterweights have been all that could be desired. We were a little afraid that the heavy counterweights on the revolving fields of the A.C. generators might cause some trouble in synchronizing these machines, but we find that this is not the case, as they couple together easily with the counterweights in any relative position. While there is a somewhat greater interchange of cur-

rent between these machines with the counterweights quartering or opposed than when in exact step, as was to be expected, this is not sufficient to cause the generators to get out of step or to produce any perceptible influence on the line.

Notwithstanding the large size of these horizontal engines and their high rotative speeds, since the application of the counterweights they have run very smoothly and with no more attention than is usually required by slower running machines. The present movement of the building (0.004 of an inch) with the counterweights of all three horizontal engines running together is scarcely noticeable.

It is interesting to observe in connection with this subject that the cost of applying the counterweights was only about two and one-half per cent. of the estimated cost of carrying the foundations to hard bottom.

Mr. F. M. Rites.—One of the most serious and intricate problems requiring the services of the expert engineer is that of vibratory forces. These forces are so varied and of such common occurrence, while their results are so difficult to predict in advance that it is impossible to provide for them or to formulate any law which shall cover them.

In my opinion, the writer of this paper is to be congratulated on solving the problem he has outlined with so little experiment, since it is sometimes almost impossible to even trace the cause of such a difficulty. The success attendant on the changes described indicate the correctness of the diagnosis of the case, which might have required a long continued course of experiment to determine the root of the difficulty.

I remember being called in consultation on a somewhat similar case at Wheeling, W. Va., where one particular engine always caused vibrations of doors and windows over a block away from the power house, while in one case a high, close, board fence perceptibly vibrated. No other engine made any such an effect, and any other engine running with this one caused a cessation of the vibration. In this case the vibration was transmitted through the air, and all that was needed was a muffler on the exhaust. An exactly similar case occurred at Allegheny, Pa., and in neither case was the exhaust powerful enough to be objectionable to the neighborhood, except in the matter of vibration.

We are all aware of the occasional vibration of office buildings by steam plants, so that most builders carefully carry piping from

the foundation, with as little contact with the walls as possible. In one case of this nature a very disagreeable vibration was felt on one floor midway of the rest and none whatever at the other floors.

The paper is exceedingly interesting, both in its description of causes and the cure of the trouble, but it seems to me the counterweight should have included in its balance the *entire* weight of all the reciprocating parts, since the entire horizontal component of the disturbing forces was to be balanced, and this would call for an even larger weight.

Mr. Geo. H. Barrus.—I wish to congratulate Professor Jacobus. In the first place, an engineer must have considerable courage to recommend the application of a weight of 3600 pounds, or nearly two tons, to the rim of a fly wheel seven feet from the center, making ninety-four revolutions per minute, as was done in this case, but the results seem to have justified these drastic measures, for, without the counterweights, the building and chimney were apparently in danger of being shaken down; while with them, the vibrations almost wholly disappeared. The diagrams given on page 540, which I understand were taken under practically the same conditions as to location, show that with the counterweights in operation there was only one-fifth as much vibration as that occurring before the weights were installed. In the second place, the Professor is to be commended for giving the Society an account of the experiences he had in overcoming the very serious difficulties mentioned. I believe that members of the Society can in no way do more for its benefit than to furnish the meetings with narratives of similar troubles which they have encountered, that fellow members may profit in their own work by the information thus obtained.

In examining the diagrams on page 540, I see that the vibrations on the switchboard platform, even under the improved conditions, are very marked. It would be interesting to know whether any attempt was made to obtain diagrams from this point before the weights were introduced, and see if the relative difference was the same as that found on the floor.

Mr. Sanford A. Moss.—A very common method of stating the counterweight problem is the following:

A counterweight is to be provided sufficient to balance the weight of all the rotating parts and a certain fraction of the weight of the reciprocating parts, all supposed concentrated at the

crank pin. The problem to be decided in each particular case then is: What fraction of the reciprocating parts is to be counterbalanced? It is customary in vertical engines to make this fraction small and in horizontal engines to make it large. Looking at the matter from this point of view, I would like to inquire what value of the fraction Professor Jacobus adopted in the case he presents.

In regard to remarks which Professor Jacobus makes on my "Connecting Rod" paper presented at the last meeting, I must admit that I have not been as clear as might be desirable in making the distinction between the effect of the weight and the effect of the inertia of that part of the connecting rod which is to be considered as a rotating part. So far as inertia effect is concerned, the fraction K'/l' is the rotating part, while so far as weight effect is concerned, the fraction a/l is the rotating part. I explicitly make this distinction, but do not bring it out very clearly in referring to the counterweight to be provided for the rotating part of the connecting rod. However, the distinction is not very important, as the two fractions do not usually differ much from each other. The importance of the distinction is also rendered insignificant, owing to the indefinite value of the fraction of the reciprocating parts which should be counterbalanced.

Mr. F. H. Ball.—This paper is very interesting to me because it is a common sense application of engineering skill. Strange as it may seem, there is a very general misapprehension of the facts in regard to the counterbalancing of the reciprocating parts of steam engines.

Comparatively few people seem to understand that there is no such thing as counterbalancing a reciprocating part by the application of a rotating mass. It does not seem to be generally known that the counterweight on the crank of a horizontal engine if applied to neutralize the horizontal thrust due to the inertia of the reciprocating parts, does not result in any better balancing of the engine, but merely transfers the horizontal thrust into vertical thrust.

In the case which Professor Jacobus had to deal with, he wisely determined that the foundation could resist vertical thrust better than horizontal thrust, and the result of his work indicates that he used good judgment.

We often hear people talk about an engine being correctly balanced for one speed and not correctly balanced for another, which of course is absurd. If an engine has unbalanced horizontal

thrusts, due to inertia of moving parts, it has them at all speeds, and the magnitude of these thrusts varies directly as the square of the speed.

It may happen that at a certain speed the number of thrusts per minute may coincide or harmonize with the time of vibration of the foundation or something connected with the engine, in which case the vibration would be more noticeable than at either a higher or lower speed, but that is due to causes entirely outside of the engine.

The consideration of this subject reminds me of an amusing incident that happened some years since: An engineer of some prominence, in a contribution to the Scientific American on the subject of balancing rotating bodies, stated that when a rotating disk is not balanced it will be found that the *light* side and not the *heavy* side is prominent, which is true if the disk is free to establish its axis through the center of gravity, but from this fact he argued that counterweight placed on a crank disk opposite the crank pin must therefore thrust toward the shaft instead of away from it, and therefore instead of reducing the thrust due to the inertia of reciprocating parts, it must augment this thrust.

An engine builder promptly attacked this conclusion in the next issue of the same publication. A little later another engineer appeared in print in defence of the original statement, which he claimed was correct and therefore the engine builder was wrong.

The amusing part of this episode was that simultaneously with the publication of the defence of the engineer's theory, there appeared in the same publication a frank statement from the first engineer to the effect that he must acknowledge his error, having found by experiment that he was wrong.

Mr. Richard H. Rice.—I desire to express my appreciation of the thorough manner in which the solution of this problem has been carried to a successful conclusion. I have used the methods described in the paper for some years in determining the angular variation of engines with fly wheels. These methods give a graphic picture of what is going on during the revolution of the engine, and from the diagrams it is easy to properly proportion the necessary weight of the fly wheel. I have had occasion to prevent vibration in fly wheels by means of weights, but in this case the weight was placed in the wheel during manufacture and found to be entirely successful. I have also had occasion to investigate the desirability of placing weights in fly wheels to enable

us to run engines in parallel, the object of this being to avoid increasing greatly the fly wheel weight, which would have involved radical reconstruction of it. In this case it appeared from computations that only very moderate improvement would be realized. In view of certain computations made by others, however, which did not agree with this, it was decided to try weights, since this could be done at slight expense. Upon trial, while the operation of the engine was improved, it was not enough improved to enable the engines to be paralleled.

The method of determining the angular displacement actually realized as described in paragraph 15 is interesting, and I regret that it cannot be used in a single unit.

The introduction of turbines into power houses containing reciprocating engines with which they must run in parallel makes it desirable to determine the angular variation of the engine, and a method of doing this would be of great value. It has been suggested that the amplitude of vibrations be measured by the waves formed on the surface of water in a vessel supported on the vibrating body. I would like to ask if this method has been tried.

Mr. E. H. Foster.—I desire to ask Professor Jacobus if they found any increase in heating of the journal on account of this weight being put in the fly wheel. I would like to know if it was necessary to use any more oil to keep the journals down to the same temperature. I would also ask how they measured the vibration at the top of the chimney. Are we to infer that a gibbet or something like that was erected to suspend a weight from above, and, if so, how long was the pendulum carrying the weight?

Prof. F. R. Hutton.—I would like to emphasize the importance of the distinction made in Mr. Rice's discussion of separating the effect of vibration in the air from the direct effect of vibration of the foundation and earth below it.

It is obvious that the seismograph devised by Professor Jacobus showed that these vibrations were the consequence of motions of the mass. As a rule, residents in the neighborhood of a power plant do not suffer from inconvenience caused by ground vibration, but the air vibration effect is very disturbing. This was brought to my attention in a little piece of work in which Professor Denton and myself were associated together.

A factory was operating a very large rotating cutter, for cutting log wood across the grain into chips to be used in the making

of dies. There was no reciprocating effect in the machine whatever, but the impact of the cutting knives in the face of the head produced a vibration accompanied by a deep note. This vibration of the air resulted in a suit being brought against the company. The experts pervaded the neighborhood with a dish of mercury and with a seismograph apparatus, but we found that very much more disturbance of both pieces of apparatus resulted from a passage of an ice cart in the street, or from a rapid ascent of a four-footed pet of the family on the house stairs, than was caused by the passage of the biggest and hardest logs through the cutting machine. The windows rattled and china on the dresser shelves.

The point which Mr. Rice has made has also come to the attention of those of us who have had vibration problems presented to us in connection with high office buildings. The pipes in a high office building attaching to the radiators and the steel structure of the building itself, behaved like the vibrating strings of a violin under the bow. A vibrating string in the physics text books is always shown with places of maximum departure from the center line and with places of no departure. The points of maximum departure are known as ventral segments, and the points of no departure are called nodes.

The tall office building has just such nodes and ventral segments under the vibration of the engine in the basement. Certain floors suffer greatly from this tendency to vibrate, while above and below will be points where no vibration can be detected.

When the period of vibration of the structure synchronizes with the pulsations of the engine, we have a vibration harmony, and the waves move together with the worst influence on the structure and the comfort of those in it. The difficulty disappears when the speed of the engine is watched so that the pulsations interfere with each other instead of synchronizing. This is the phenomena so familiar on high speed boats.

Mr. Oberlin Smith.—Does the steam turbine have any alleviating effect in this trouble from vibration in tall buildings?

Mr. F. R. Hutton.—Mr. Smith's question is very pertinent. The high speed turbine produces an air vibration, but not a ground tremble. The buzz or hum of the turbine would set things in vibration through the air if the noise were allowed to get out of the engine room.

The trick is to fit the engine room with double doors and

weather strips, so that the noise or buzz is entirely confined within its walls.

Professor Jacobus.—Mr. Barrus says that it seems quite enterprising to place the large counterweights in the engines. This reminds me that I should have said more about Mr. Bushnell, the chief engineer of the company, in presenting the paper. As stated in the paper, Mr. Bushnell worked with me in designing the counterweights, but I should also have said that it was his enterprise in being willing to try what I suggested, that made it possible to employ the large weights.

Mr. Rites says that it seems to him that the counterweight should have included in its balance the entire weight of the reciprocating parts. We appreciated that this was so, as far as the shaking forces were concerned. For such a full counterweight, the mass in the fly wheel should have weighed 3960 pounds, but we cut it down to 3600 pounds because this gave very nearly as good results as the larger weight, and we considered it advisable to keep slightly within the theoretical limit. There was a point where by adding to the weight there was but little difference in the shaking force; and we recommended a weight which we thought, when all things were considered, would give the best results.

Mr. Barrus asks if we made any attempt to secure diagrams showing the vibrations of the switchboard platform before the counterweights were applied. We did not do this because the switchboard platform was not erected when we first measured the shake. We secured such diagrams later on because the switchboard platform vibrated more than any other point in the building. It seemed that the side wall of the building, to which the switchboard platform was attached, was in tune with the vibrations of the engines, and as a matter of interest we registered the amount which it vibrated. The measurements which give the amount of improvement through the application of the counterweights are those which show the shaking of the floor.

Mr. Moss has spoken on the theoretical side of the problem, and has referred to the note which I made on his paper. Mr. Moss' paper is most carefully prepared from a mathematical standpoint, and the equations which he gives are correct. It was only in the discussion of his formula that his remarks may be misleading.

In balancing an engine, if we wish to obtain the shaking forces correctly, we have only to assume that mass is divided between

the crank and wrist pins in proportion to the gravity components of the weight of the rod at these points. This was brought out in the papers, already referred to, by Naval Constructor Taylor and by Admiral Melville. The distribution of connecting rod weight proposed by Mr. Moss does not give as correct results for the shaking forces as this simple method, but, on the other hand, it gives more correct results if used in determining the power transmitted by the engine.

Now to come to the questions asked by Mr. Foster regarding the heating of the journals on account of the weights being placed in the fly wheel, and the measurement of the vibration of the chimney.

Fortunately the journals on the original machines were of ample size so that with the counterweights added to the wheels the pressures were not excessive. There has been no trouble experienced since adding the counterweights in the heating of the journals.

The vibration of the chimney was measured by the same instrument which we employed for the rest of the work. We suspended the weight used in connection with it by means of a couple of poles, the weight, weighing perhaps 400 pounds, being held by about fifteen feet of wire. We were very careful to get the time of swaying of the weight and to make sure that it did not come in harmony with the vibrations caused by the engines. It was found to be entirely out of phase with the chimney, so that the results obtained with the instrument on the chimney are reliable; but, as I have said, the maximum vibration of the chimney which occurred when we slowed down the engine, so that the chimney came in tune with the strokes of the engine, was beyond the range of the instrument.

I wish to thank Mr. Ball, Mr. Rice and Professor Hutton for their kindly discussions and the additional data which they have presented.

Mr. Baker.—Engineers engaged in locomotive design have become familiar with the balancing of reciprocating parts, and it is well understood that whatever weight is put into the driving wheels to counterbalance the horizontal motion of the reciprocating parts is entirely unbalanced vertically and produces, consequently, the so-called hammer-blow upon the rail.

I should be interested to learn whether the vertical effect due to these counterbalances at the lowest point of their revolution

was computed, and whether the journal pressure referred to in the paper as not in excess of good practice was that due to the static load or included the maximum vertical effect of these counterweights with the engine at full speed.

Professor Jacobus.—By adding the counterweights we changed the vertical shaking forces of the horizontal engines to about those which would exist in vertical engines of the same size. That is, we converted the horizontal shaking forces to vertical shaking forces. As already stated, the journal pressures were not excessive. We worked out the centrifugal force of the fly wheel counterweight and determined the pressure due to it, but even including this force, which acted intermittently on the upper and lower bearings, the pressure was not a dangerous one. When we consider the friction of the journals, the most important element is the dead weight because this acts continuously.

No. 1082.*

FORMATION OF ANCHOR ICE AND PRECISE TEMPERATURE MEASUREMENTS.

BY DR. HOWARD T. BARNES, MCGILL UNIVERSITY, MONTREAL, CANADA.
(Non-member of the Society.)

1. My attention was first directed to this very important problem by the Harbor Commissioners of Montreal in 1896. The Chief Engineer of the Harbor Commissioners, Mr. Kennedy, was a member of the Flood Commission to investigate the very serious floods that visited the city of Montreal from time to time, and he was appointed with other gentlemen to go into this question with a view of alleviating the disastrous effects. During the investigation of the Flood Commission efforts were made to study the slush ice, which is called in Canada frazil ice, and to connect the formation of this remarkable ice with the temperature changes in the water. A great many results were obtained showing that these temperatures varied a good deal. Calling for observations on this point many independent observers throughout the country took temperature readings during the winter time with instruments at hand and sent their results in to the commission. The commission also took a series of measurements of the temperature of the water and they thought they detected small differences of temperature during such times as the ice was most severe. But they were not sure of their results on account of the difficulties encountered in making the measurements. Other observers were not, however, so careful and they sent in results showing apparently that the temperature of the water varied from five to six degrees below the freezing point. Now, it was easy enough for the commissioners to explain why these very low results were obtained when they received details in regard to the manner in which the measurements were carried out. Usually it was a mer-

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cury thermometer thrown in the water and held up in the air to read. Mr. Sproule, the Assistant Engineer of the Commission, devised a tin case for his thermometer in which he could entrap a small amount of the water and thereby have a longer time to read the thermometer after drawing it out of the water. The thermometers were not delicate enough, and as Mr. Sproule told me, when the question came up in 1896, that during the time of the commission and later it was impossible for him to say definitely that any deviations from the freezing point took place, his thermometer being graduated to degrees, and such variations that would occur beyond this point were not recorded. That is why the commission came to McGill University for more refined measurements. They wished, if possible, to connect the temperature of the water with the formation of frazil ice.

2. We devised then an instrument for doing this which depended on the change in the electrical resistance of platinum wire. Now such instruments have been made before and have been exceedingly useful for all classes of temperature measurements; but we devised the most delicate differential thermometer that probably has ever been made. This thermometer has a degree which is eight inches long, and it is possible to estimate by means of it to the ten-thousandth part of a degree. Later I shall speak a little more about the practical details of the measurements.

3. My first object is to take up the problem of ice formation and so lead on to the measurements, and connect these measurements with the loss of heat from water which governs the formation of ice in the laboratory or in the actual conditions of nature, which problem is not an easy one. The varied conditions in which we meet river ice make it almost impossible to establish a definite set of rules by which we can say how much ice will be formed or what the ice will do. The weather itself varies from year to year and those conditions which will hold one year will be entirely different the next. A long and continuous study of each problem of engineering should be made before anything definite can be said in regard to the effects of river ice.

4. Now I want first to run over the laws of heat which I shall have to refer to, and I do this for the sake of clearness. I will do this very briefly, somewhat in the form of a review, but if we are to get a connected and scientific knowledge of the subject we must review first of all our elementary physical knowledge of heat transmission, for, after all, the problem is a problem of heat

transmission. The extraction of heat from the water causes ice to form. How is this heat abstracted and how does water gain heat again?

5. Now the three recognized methods of transmission of heat are *convection*, *conduction* and *radiation*. This does not mean that we have three different kinds of heat, that one kind of heat is lost by conduction, one by convection and one by radiation. It means three processes are operative.

6. The first method, convection, we may briefly define as the transmission of heat by the movement of matter itself. You have illustrations of this in the ocean currents and trade winds which are so important in tempering the earth's climate. It depends entirely on the fact that matter when heated expands and becomes less dense. Therefore, a heated portion of matter surrounded by cooler matter will be displaced by the cooler matter as a light object is displaced by water. So we have heated matter rising when surrounded by cooler matter. We have one exception to this in water at 4 degrees Cent. As an illustration of this showing the maximum density of water we have twelve thermometers of equal bulbs containing equal weights of water. One bulb is placed at zero and the others in successive degrees to twelve. At four degrees we have the maximum density or the liquid occupying the least volume. Below that point to zero, water expands. Above that point, as high as we can go, water expands; hence we see that in the case of water the process of convection will cause a current of warmer water to rise until we get to four degrees, and then the reverse process will take place. Below four degrees, as the water becomes cool, the warmer layers sink to the bottom. Now, that has a very important influence, as we shall see presently on ice formation in lakes and quiet water.

7. The next process of heat transmission is conduction. By conduction we mean the transmission of heat from point to point in the body. Heat is a form of energy; it is a measure of the vibration of the molecules of a body. When the molecules at one point become warm or vibrate more energetically these vibrations are conveyed to neighboring molecules.

8. Conduction only affects the ice problem in so far as it causes a thickening of the sheets of ice over a river or lake. We can measure the conduction and we know it in definite units for ice and for water, and in any problem in which conduction enters we can calculate with a fair degree of exactness the rate at which

ice will form; otherwise, conduction plays a very small part in river ice formation.

9. The next process is that of radiation, or the transmission of heat by ether waves. All bodies radiate energy by ether waves which travel out in all directions in straight lines with a velocity of 187,000 miles a second. They are, in fact, merely a continuation of the ether waves, which are capable of affecting the optic nerve, and which we call light, and we know some of these light waves possess heating qualities.

10. These waves extend for a long distance beyond the visible spectrum and they are being investigated and the limit pushed further and further down. All light waves travel with the same velocity and hence the length of the waves and the number of vibrations of the waves must vary correspondingly. The longer the wave the slower the vibration. Now we know that this is one fact that distinguishes the heat waves from the light waves. We can extend down for a long distance, almost to connect with the electrical waves which we are able to produce, but we have not yet bridged over a long gap which exists. When we come to examine the whole spectrum of ether waves we pass up from the electrical into shorter waves which possess heating qualities. We pass on to shorter waves still light; we pass beyond these into waves which are still shorter, ultra violet; we pass on further, and we come where—perhaps to the X-ray, for the latest ideas in regard to these X-rays are that they are very short waves of ether.

11. We can prove in a wonderful way that the heat waves are similar to light in that they may be reflected, refracted and polarized.

12. Now the problem of heat transmission by radiation is the most difficult. We know the least about it. A great deal has been done recently in unearthing the laws of radiation from black bodies, and in applying our knowledge of radiation to problems we have, first of all, to make a calculation of what it would be if the object were a black body. Then we must make estimates. Every kind of surface depending on its roughness and its temperature will send out or absorb different kinds of waves. In general, the higher the temperature of the body the more of the light waves, or shorter waves are sent out and the fewer the longer waves. The lower the temperature the fewer of the shorter waves will be sent out and the more of the long. Finally we get bodies which are not luminous at all, and then if we take bodies

still colder we find that they emit longer and longer rays the colder they become. Now the difficulty in applying the laws of radiation to practical engineering problems is that we do not know enough about surface conditions or radiation from cold bodies.

13. Bodies differ a great deal in their power of absorbing rays. Some bodies transmit heat waves; some do not. We have, for example, in colored glasses, very beautiful illustrations of how some bodies transmit light waves and some do not. Some of the rays are absorbed and some can go through. Tyndall has shown us that for heat rays water and ice are very opaque, and he has pointed out that of the longer rays which are emitted from an object heated one hundred degrees above its surroundings, the surroundings being the ordinary temperature of the room, these would be entirely stopped by a mass of water or a mass of ice. Ice is a little more opaque to heat rays than water. From the heat rays that are produced from an Argand burner only 11 per cent. penetrate a small thickness of water. The penetrating rays, however, manage to pass through further and thicker layers of water. The filtering process is very complete, but the rays which penetrate pass on through the water unabsorbed.

14. This property of selective absorption which is shown by many substances is shown particularly for water, and it has only recently been proved experimentally that a large absorption band exists in the case of water for the longer heat rays. An absorption band is a band in the spectrum where the rays have been removed. Now to make the explanation clear, consider some examples. It is an exceedingly important point that I wish to bring out, and one which is to determine whether the long heat rays can penetrate water. Now first of all consider the spectrum. If we can study the light rays in regard to their absorption in bodies then we can apply our knowledge to heat rays. We find that each vapor or gas has its characteristic spectrum. A good illustration is afforded in the absorption spectrum of iodine or bromine vapor. The light transmitted through the vapor when refracted through a prism shows dark bands together with characteristic colors. These dark bands show where the light at that particular wave has been filtered out and been absorbed in passing through the vapor.

Often the dark patches or absorption bands extend for a long distance through the spectrum.

15. Tyndall showed that the long rays from copper heated 100

degrees are all stopped by water. Rubens and Aschkinass have shown that there is a dark band (an absorption band) in the heat spectrum of water, and therefore there may exist heat rays beyond this absorption band for the longer waves which will penetrate the water. There is nothing to prevent our making that assumption. I will admit that it is yet to be proved that such exists, but that such could exist seems entirely plausible. Take the bed of a river which is at the freezing point radiating heat off into space. The waves of heat which will pass off will be very long and they very probably consist of waves beyond the absorption band discovered by Rubens and Aschkinass.

16. Having run over the various points in connection with heat transmission I want to dwell for a short time on the physical properties of ice and water. Let us take ice first and briefly review its characteristic properties. We know that ice is very plastic; it can be molded into various shapes. We have examples of ice which has been pressed between heavy molds to form perfect spheres; and ice that is pressed in a cylinder may be forced out of a narrow opening. Ice, however, does not exhibit the characteristic properties of viscosity. This was pointed out by Tyndall. An illustration of this is the glacier action. A glacier in moving from a narrow passage to a wide passage cracks instead of adapting itself to the wider passage. A glacier moving down over a very gentle slope will crack on lines at right angles, showing that ice is exceedingly brittle. Therefore, we cannot say that ice possesses viscosity to any great extent, although it can be molded under pressure and made to run under pressure. Now the phenomena which enables it to give under pressure is probably due to another physical property of ice, namely regelation. We know that pressure sufficiently great will melt ice. One hundred and fifty atmospheres pressure will lower the freezing point of water and ice 1 degree Cent. It does this by squeezing some of the ice back into water; the temperature is lowered by the melting of the ice and the absorption of the heat. Regelation accounts for a great many ice phenomena of nature. A very beautiful example of regelation is Bottomley's experiment. It is a block of ice being cut through by a wire attached to a weight. I dare say many of you have seen the experiment. The pressure of the wire at the point of contact melts the ice. The water runs around the wire and finds itself freed from the pressure and at a temperature lower than its freezing point; it freezes again above the wire.

In this way the wire slowly creeps through the ice leaving it as strong after the wire passes as it was before. By means of pressure we can freeze blocks of ice together—two pieces of ice brought together and pressed will stick together. A slight film of water is formed on the touching surface and immediately the pressure is relieved the water in contact with the ice surface finds itself below the normal freezing temperature at once solidifies, cementing the two blocks together.

17. There is another property which ice possesses and which has quite recently been measured. That is, it possesses the power of growing by itself; ice in contact with water exactly at the freezing point apparently goes on growing. You have all observed the beautiful growth of a crystal of alum or a crystal of sugar in a saturated solution of the alum or sugar, and how it gradually increases in size. Apparently ice does the same thing, only very slowly. As I shall explain to you presently water possesses ice in solution. The rate of growth of the ice crystals has been measured. If we allow the natural rate of growth to go on it would form an inch in about two and one half years. So it does not form a very important factor in practical engineering problems. It forms an important factor, however, in this respect, that it helps to cement ice crystals together. It was Faraday who first showed that if we had some cracked ice floating in a vessel by taking one of the pieces of ice and bringing it in contact with another piece of ice it would stick to it; no pressure required, merely contact; and would draw it along. So he was able to get a whole train of six or eight pieces of ice in a vessel all being led by the first piece. This process of growth shows why these particles cemented themselves together when touched. For at the point of contact we have two ice surfaces with zero water between. The ice would be growing all the time and hence the two pieces in contact would at once adhere. A pressure is therefore not required to cause ice to freeze together.

18. Now of the physical properties of water we have the most important, *i.e.*, maximum density. We then have the facts that water is opaque to heat rays and that it allows only a few of the heat rays to pass through. We have a further and quite an important physical property, that below 40 degrees Cent. or about 97 degrees Fahr. water commences to form ice; this ice remaining in the solution until the freezing point is reached. The proportion of ice increasing as the temperature falls and at the freezing

point we have actually a saturated solution of ice and water. It explains to us the maximum density of water and the minimum specific heat.

19. The curve showing the variation of the specific heat of water drops very rapidly from 0 degrees Cent. and passes through a minimum at 40 degrees. Now that is apparently due to the fact that besides the heat absorbed in the natural expansion of the water we have the heat absorbed in transforming those ice molecules into water molecules. That is as the temperature rises, the amount of heat that would be absorbed by the ice becomes less and less. At 40 degrees Cent. we have an exact balancing. From 40 degrees upwards we have the water existing as a pure liquid.

20. Now I think I have taken up the chief points in regard to the physical properties of ice and water with the exception of the great change in volume that takes place when water freezes, which has such an important influence in river ice formation. There is an increase of about 9 per cent. in volume, and this is exceedingly important inasmuch as the ice as soon as it is formed comes to the surface or remains on the surface. Take for example a sheet of quiet water which is cooled lower and lower by the advent of winter weather. The surface becomes cool; the cooler layers fall to the bottom until 4 degrees is reached. Then the warmer layers stay at the bottom and the cooler layers remain on top. As soon as the surface gets to the freezing point ice forms. This ice spreads over the surface, usually from the banks towards the center, and gradually increases in thickness by conduction.

21. To the convection of air currents we probably owe the fact that the ice spreads from the edges to the center, because the water heats the air and it therefore rises from the center and colder air sweeps in from the banks to the middle. We therefore have the water along the edges always in contact with colder air than at the center. It has been considered that conduction plays probably the most important part in this phenomena. But the conductions of heat through the banks of the river is slow indeed. It is so slow as to be practically negligible.

22. Let us consider what goes on in the case of a river which is flowing too rapidly to form a surface of ice? Naturally we will have instead of the flat surface ice ice of finer quality formed. Each little piece of ice produced by the cooling of air on the sur-

face is carried down by the currents and submerged, rising to the top again as the currents carry it above. Not forming the protecting layer of ice the flowing river loses a very much greater amount of heat and the fine ice is produced by agitation in very great quantities. This is to be expected if we stop for a moment to consider river ice formation. How is this heat taken away from the water? We have the three governing processes at work. Conduction is probably negligible; convection by means of air currents and radiation from the water which is warm compared to the air above it.

23. Now you know that these fine crystals of ice which are formed and which are called frazil crystals, are swept down with the current and are carried by the current for long distances from open water under the surface ice. It was one of the discoveries of the Flood Commission of Montreal that this ice was found many miles below the bordage ice at the foot of the Lachine Rapids, where the greater portion of this ice is formed. An inspection of the many diagrams of the Flood Commission for various sections of the St. Lawrence River below the city of Montreal, shows immense accumulation of frazil or what the engineers have called also sponge ice. It was found on examining this ice that it was very spongy. In determining the depth of the ice a leaded weight could be lowered through by slight pressure from above, and it was found that this ice accumulated under the surface ice to depths ranging from 30 to 35 feet. That it acted as a filter to the water which penetrated it was shown by the accumulation of a great deal of matter from the water, just as a sponge accumulates slime from water. The great bulk of this ice which they found was manufactured in the Lachine Rapids, and in the reaches of open water extending about six miles from the ice sheet on Lake St. Louis. They found in many places below the rapids that there was more ice than water. The percentage of this ice became less as they obtained their suctions further and further below. Until finally, some twelve miles below, the frazil was exceedingly scarce. In general, wherever an open river flows unprotected, quantities of this frazil ice are formed by radiation and by convection of air currents and wind agitation. These crystals of ice vary in size, depending upon the agitation of the water. The more rapid the agitation the finer they are, and the further they travel the larger they grow.

24. There is another form of open water ice which we must

consider and which is primarily the subject of this paper, namely, *anchor ice*. Anchor ice and frazil ice are often mistaken for the same thing; as a matter of fact we must distinctly draw a line betwen the two. Frazil ice is ice formed in the water itself by surface agitation and other causes. Anchor ice is ice that grows on the bed of the river, on the rocks, on anything in the bed of the river to which ice may be attached.

25. There is still a difference of opinion as to how this anchor ice grows; how it first forms; but I think when we consider the amount of heat which is lost from the surface of a river by radiation and the fact that much of this heat can penetrate through the water that we must admit that radiation plays an important part. When we consider further that this anchor ice is formed only at night when the radiation from the earth is a maximum, that as a rule it never grows under a bridge or under any object which would reflect the heat rays back again into the water, and when we consider that anchor ice apparently grows more rapidly on dark rocks than on light rocks then we must admit there is strong suspicion that radiation plays an important part in its formation. That anchor ice is built up as well by the freezing to the bottom of frazil crystals goes without saying. It could not do otherwise. These surface currents carry down the fine pieces of ice, which passing across the layer of anchor ice which may have formed there by radiation become attached, and we have then great thicknesses built up in a single night. Anchor ice is formed partly by radiation and partly by the sticking to the bottom of these frazil crystals. Anchor ice, during a severe spell of weather, when it is cloudy by day and clear by night, attains a thickness of several feet, such that a pole or a stick placed on the ice will sink into it to considerable depths. When the sun rises it has been noted for a long time that these masses rise to the surface. The boatmen are very charry about going across an open channel when the sun is bright, for these masses come up in great quantities. As soon as the sun gets at a sufficient angle so that its rays penetrate the water, they pass through the water and ice, warm the bed of the river and warm the water in contact with the ice. The ice rises with a characteristic hissing noise several feet above the water and then falls back turning white as the water drains out of it. It floats low in the water because it is saturated and waterlogged just as sawdust will float lower in the water than an equal mass of solid wood. The fact that these

anchor ice masses float lower in the water than a similar weight of packed ice would, has given rise to the opinion that anchor ice was ice denser than normal and therefore held down by its own weight. There is no evidence whatever to show that ice of a greater density than normal exists and we therefore disregard that idea as being entirely untenable. Ice at one particular density is formed and only one. The reason of anchor ice being fastened to the bottom is that it is actually frozen there. As evidence of its buoyancy, large masses of rock are sometimes brought up with the anchor ice and carried down the river. Anchor ice will carry large boulders which fill up the channel of the river, and that is one thing the Montreal harbor commissioners have to contend with in the harbor there. They have annually to remove a large amount of rock and boulder carried down by the anchor ice. This anchor ice when it comes up becomes disintegrated and is carried down with the currents, thus aiding the frazil in causing the damming up of the channel.

26. Now it was the problem set me by the Harbor Commissioners to determine if possible what were the relations between the temperature of the water, if any, and the formation of anchor ice on the bottom, and the peculiar agglomeration of frazil crystals that takes place during very cold weather. It is a great difficulty that the engineer has to contend with in clear water with wind agitation that these little harmless particles grow in size very rapidly and form hard masses to which other crystals immediately adhere. Under certain atmospheric conditions ice agglomerates very rapidly and objects placed in the water become rapidly coated with it. It is a common phenomenon that if one lowers a bit of iron or metal into the water when the water is in this state that the iron will freeze at once to the bottom or to whatever ice it comes in contact with. Mr. Keefer, who has done so much in studying ice formation, has observed in going across the St. Lawrence River during an exceedingly cold day when the water was very dull and appeared to be completely laden with frazil crystals, that he could feel the resistance to the motion of his paddle as they attached themselves to it, sticking out at right angles like long needles.

27. We designed a suitable form of resistance thermometer and undertook in the winter of 1896 to determine if possible any variation in temperature from the freezing point under the surface ice. Accordingly, a shanty was provided for the instruments

in a suitable spot opposite Montreal, and in February of that year the instruments were taken out and observations commenced. We wanted to know how much the river differed from the freezing point. The method of observation which was adopted was a differential one, determining the temperature of the river by a difference from a carefully prepared mixture of ice and water. A very sensitive galvanometer was used and any changes were at once detected by either a change in the galvanometer or a change in a suitable measuring wire. The wire having a scale, as I mentioned before, of 8 inches to the degree centigrade. Now the difficulties encountered were very great, and I may say although about five weeks were occupied in these measurements only half a dozen measurements could be obtained, for troubles arose from the breaking of the thermometer; from the breaking of the galvanometer owing to the roughness of the shanty, and from the varying magnetic field due to the heating apparatus.

28. The results of these observations have been published, and it was found during the entire time, with a variation of air temperature from 28 below to 40 above, the river under the surface ice never varied as much as one one-hundredth of a degree from the freezing point. That during the entire time of the observation no deviation as much as that could be measured and as a rule the variations were within a few thousandths of a degree. A severe spell of cold weather would cause a slight drop in temperature and vice versa.

29. The tube of the thermometer passed out through the shanty to the river, from a shelf where rested the galvanometer and measuring instruments, which I may say were of priceless value as they were unique of their kind. The bulb of the thermometer was encased in lead 6 inches long connected with wires encased in lead 100 feet long which passed up into the shanty. First of all an exact balance was taken between the resistance of the two bulbs when both were immersed in the ice and water mixture in the shanty. The watchman then carried the bulb directly out and put it into the river, and after a few moments another balance was obtained. It was possible to obtain at once a change in reading giving by reduction of the temperature.

30. The following winter, observations were made at the Lachine Rapids because it was considered that possibly the water under the ice opposite the city came to the freezing point because it flowed so far under the surface sheet of ice. The shanty pro-

vided was smaller again than the one we had before and there we had further difficulties. It was placed on the coffer dam of the Lachine Hydraulic Company, then under construction, and just at this point the water was running at a very high velocity. This sheet of water is entirely open and flows rapidly from a point about five miles above the dam and sweeps around this curve with a tremendous velocity and a great deal of surface agitation.

Fig. 202 shows the position of the observation shanty. The dam is seen extending out to the corner where the shanty was placed;

FIG. 202.—OBSERVATION SHANTY, LACHINE, WHERE TEMPERATURE MEASUREMENTS WERE MADE OF THE WATER.

the river runs down here to the rapids proper which are just at this point.

31. Fig. 203 shows the agitation of the water as it passes down around the ice which grew out from the edge of the dam. The watchman with the thermometer in his hands is seen prepared to put it in the water. This was a good spot to observe the river temperatures, because from this point, for say five miles, the river being a mile wide and fairly shallow, the water was cooled by air currents. Therefore one would expect that if water ever got cooled much below the freezing point it would have a chance in this case. The water was only a few feet deep near the dam but became rapidly deeper as we got out. The thermometer tube could be bent and placed at quite a distance

under the ice edge so as to be protected from radiation. Measurements were carried out for about six weeks in this position and we got slightly greater variations than under the surface ice the winter before, but as far as the lowering of temperature was concerned it could be limited within one one-hundredth of a degree centigrade; and the greatest lowering of temperature observed during a very severe day was six thousandths of a degree. Photographs taken of the agitated water on a mild day and on a cold day show distinctly a difference. On the mild day the water looks softer and less stiff, but on the severe day it looks

FIG. 203.—ATTENDANT ABOUT TO PLACE THERMOMETER IN WATER.

harder as though the spray was at once frozen and fell back as ice. Great masses of anchor ice would rise near me and go floating down past the shanty. They look like a great mass of solid ice, but if one should attempt to test their soundness one would go right through. We have witnessed many hundreds of these rising to the surface after the sun gets up.

32. Table I. shows the results obtained in these measurements, with water temperatures, clearness of sky, wind and miles per hour between the rapidly flowing current and the quiet back water just below the point of the dam. As the current was swept around the dam, some of the water came back and formed an eddy.

On the third, the sky was clear and the weather was fine.

On the fifth we have + 32 degrees and — 3 degrees; wind much lighter; and a temperature above freezing of 0.02 of a de-

gree. The reason of that is the great deal of sunshine, 88 per cent.

The seventh was a rainy day. We obtained quite a rise in temperature; nearly 0.02 degree above freezing.

On the following day it was raining also. We find the influence of the rain making itself felt in the river.

TABLE I.

Date	Air Temp. Fahrenheit Degrees.		Sunshine in per cent. if possible.	Sky.	Wind in Miles per hour.	Difference from Freezing Point in Degree Centigrade.	
	Max.	Min.				Current.	Quiet Water
Feb. 3d....	+ 20	+ 10	89	Clear.	28.6	0	.....
" 5th...	+ 32	— 3	88	"	9.7	.....	+ 0.0215
" 7th...	+ 38	+ 27	00	Raining.	12.1	+ 0.0197	+ 0.0182
" 8th...	+ 34	+ 28	00	"	16.3	+ 0.0547	+ 0.0415
" 11th...	+ 18	— 3	94	Clear.	6.5	+ 0.0137	+ 0.0151
" 12th...	+ 11	— 2	00	Stormy.	21.1	— 0.0065	— 0.0068
" 13th...	+ 17	0	100	Clear.	15.1	.....	+ 0.0186
" 15th...	+ 33	+ 24	67	Clear to cloudy.	17.6	+ 0.0280	+ 0.0423
March 1st ...	+ 22	— 10	25	" " "	16.8	.....	0

During these two days of rain the river was fairly well cleared out of floating ice; very little ice could be seen. On the twelfth a very stormy day with a high wind, 21.1 miles an hour; no sunshine; a mean temperature of about 7 degrees, we found a lowering of the temperature of six to seven thousandths of a degree and the measurements corresponded very closely for the current and quiet water. During this day the river was filled with fine particles of frazil and a great deal of ice was formed. I might say that as one of these measurements was being made the thermometer bulb was resting on the ice on the bottom and when I came to remove it I found it frozen solid there, and it was with great difficulty that I was able to remove it without injury.

The next observation on the thirteenth was made with a clear sky as you see here again the temperature is above zero.

The next observation on the fifteenth was similar, with a fairly high temperature; 33 degrees maximum and 24 degrees minimum. We get the influence of the sun again.

This table illustrates one thing very clearly, that the margin between the disintegration of the ice and the formation of the ice is exceedingly narrow. That it is only a few thousandths of a degree. These high temperatures are all influenced by the sun's rays being absorbed in the water.

33. Table II. is reproduced to show the effect of the sun's rays in the water.

TABLE II.

Date.	Sunshine in %.	Locality.	Diff. from 0° C.	Air Temp.
Feb. 13th.	100	5 ft. from surface (bottom).	+ .0186	+ 16° Fahr.
"	"	1 " " "	+ .0474	"
"	"	8 in. " "	+ .0741	"
"	"	Shallow current.	+ .0461	"
Feb. 15th.	67	3 ft. from surface.	+ .0423	30° Sky Clear.
"	"	8 in. " "	+ .0819	" "
"	"	Just under edge ice.	+ .0292	" "
"	"	Deep current.	+ .0280	" "
"	"	Bottom of back water, sun clouded.	+ .0112	" Cloudy.

On the 13th of February, five feet from the surface, we got a measurement 0.0186 degrees Cent., the air temperature being 16 degrees Fahr., at two feet it was 0.0474 degrees Cent. Although the air just above it was at 16 degrees Fahr., on the fifteenth similar measurements were made. During the time of that measurement, which is the same locality as before, the deep current and quiet water, 3 feet from the surface, were both 0.0423 degrees. The sun clouded over and the effect was at once seen in that the temperature had dropped from 0.0423 degrees Cent. to 0.0112 degrees Cent. Now if the sun had remained cloudy and the air cool the water would have dropped to the freezing point and a little below. The great influence which the sun has in warming the water is illustrated by the fact that anchor ice never forms and river ice never has any serious effects while the sun is shining.

34. Fig. 204 is a diagram which illustrates the narrow margin of temperature with which we have to deal in river ice formation. The horizontal axis represents the per cent. of ice in the water. On the vertical scale we will put the difference of temperature, plus above and minus below. Starting from the point representing pure water and carrying off a fixed and definite amount of heat per second, we would find that the temperature of the water would drop to about 0.0104 degrees, and ice would immediately commence to form. As the percentage of ice became larger the temperature approaches the true zero degrees Cent. Conversely, suppose we start with 50 degrees of ice and continuously apply heat to melt the ice, the temperature will rise a little above the freezing point, and as the percentage of ice grows less the tem-

perature rises higher. In the river the condition would be represented by the dotted region. That is, we would never get beyond a certain percentage of ice in the open water. We might get 20 per cent., but not exceeding that in the open water. Here we would find the temperature fluctuating between limits represented

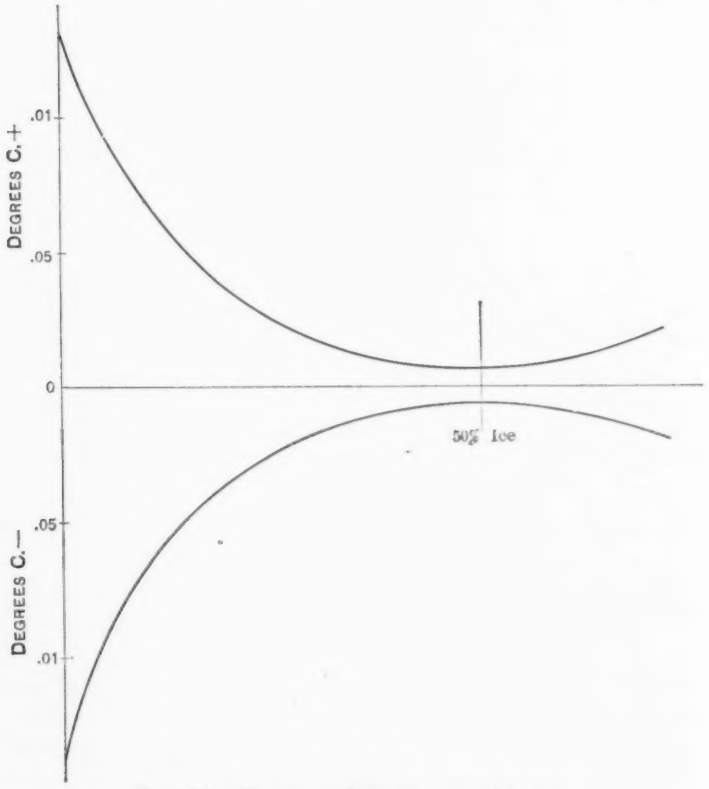

FIG. 204.—FREEZING TEMPERATURE DIAGRAM.

by that dotted line depending on the river losing or gaining heat. When the air temperature is sufficient to cause a rapid abstraction of heat the process of freezing goes on and the temperature falls slightly below the freezing point. The reason for this is probably that ice only crystallizes at a fixed rate. If we abstract heat rapidly enough and the water cannot freeze rapidly enough to keep the temperature up to the freezing point the only thing that can happen is that the temperature must drop a little below.

When the reverse process takes place and heat is applied to the mass of water, the ice will not be able to melt fast enough to keep the water cool, and the mass will in consequence be a little above the freezing point. The less the amount of ice present the more the temperature will rise or the lower the temperature will fall for any given rate of gain or abstraction of heat.

35. I hope I have made it clear from the diagram that what we have always considered to be a fixed and definite temperature, namely, zero centigrade, or 32 degrees Fahr., is really not absolutely fixed. It varies a little one way or the other, depending upon whether the mixture is receiving heat rapidly or losing it rapidly.

I have made measurements of these differences up to 20 per cent. of ice and the method I adopted was a simple one. I took a quantity of water and cooled it rapidly by bubbling liquid air through it. This liquid air carried off a definite and regular supply of heat from the water and the agitation of the water caused fine crystals of ice to form, looking in every way like frazil crystals. Just before they formed the water sank to —0.014 degrees. As soon as the ice began to form the temperature rose rapidly. On adding fine crystals of pure snow to the water to increase the percentage of ice the temperature rose nearer to the freezing point, —0.006 degrees Cent.

36. Now as to the rate at which surface ice will form, I would say that we were intending to try some experiments this winter to determine the answer to that question. I have recently examined into the data on the matter and I think a very simple formula based on mathematical calculation can be obtained to represent the rate at which ice will thicken; and I should like this to be tested experimentally some time and so put on a firm basis. But we can write this expression here as connecting the time T at which ice will grow with the thickness. We find that $T = \frac{Lsd}{K\theta}\left(1 + \frac{d}{2}\right)$ where L = latent heat in calories (8.0); s = density of ice (0.9166); K is thermal conductivity (0.0057 cal. per square centimeter per sec. per deg. diff.); θ = temperature difference between ice at 32 degrees and the air temperature; d is the thickness of the ice sheet in centimeters which will grow in a time T seconds.

This gives the thickness of the surface sheet with water temperature at 32 and air temperature 3 degrees below zero Fahr.;

as 3 feet in 44 days. Starting with water without any ice on it at all it gives a thickness of ice of 3 inches in eight hours. I think in most places this would represent fairly closely the rate of the growth.

37. I wish now to say something in regard to the measurement of temperature with these electrical thermometers. I have worked for a great many years with them, and I may say that they are capable of the greatest possible refinement. I have here a number of differential thermometers which I used in my experiments and in verifying my conclusions in regard to the natural growth of ice, I used a pair of differential thermometers reading temperatures to 0.0001 degrees Cent.

An examination of these thermometers will show that they consist of a coil of platinum wire. Their limit of accuracy depends upon the limit to which we can measure resistance. When we come to measure hundredths of a degree the mercury thermometer becomes troublesome; when we come to measure thousandths of a degree the mercury thermometer is almost impossible; when we come to measure ten thousandths of a degree we cannot use the mercury thermometer at all. To measure ten thousandths of a degree with platinum thermometers difficulties are of course met with. Some observers claim to have obtained measurements to a millionth of a degree. I do not attach much weight to that because I do not think anybody knows what a millionth of a degree means.

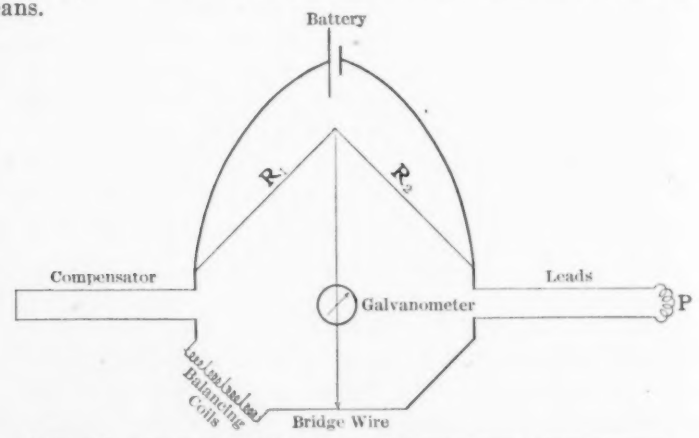

FIG. 205.—DIAGRAM OF PLATINUM THERMOMETER CONNECTIONS.

38. The diagram of connections for the platinum thermometer is shown here (Fig. 205). Two ratio coils R_1, R_2, one arm with

the coil attached, P, together with the necessary leads, a fourth arm containing the compensating leads and balancing resistance, forms the Wheatstones bridge arrangement. The compensating leads are equal in length to the thermometer leads and run side by side with them.

In the differential arrangement, Fig. 206, we have a double set of connections. The compensating leads of thermometer A are connected in series with thermometer B, and the compensating leads for thermometer B are connected in series with thermometer A. Thermometers A and B, forming the differential pair, are con-

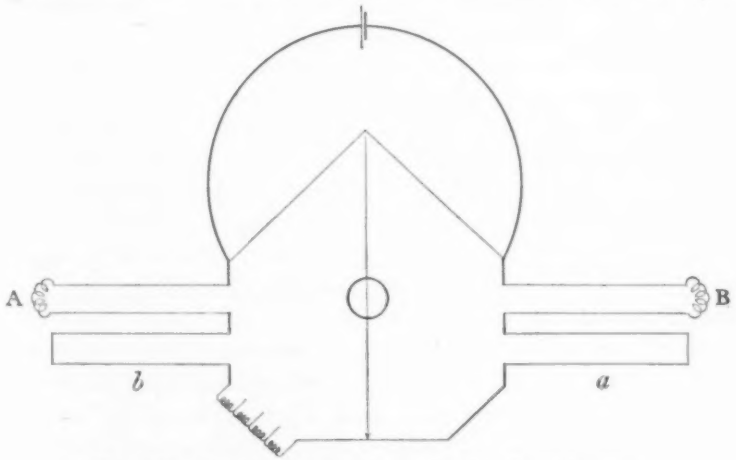

FIG. 206.—DIFFERENTIAL THERMOMETER CONNECTIONS.

nected to opposite arms of the bridge. There is a bridge wire on which small variations can be measured and if the reading goes beyond the bridge wire compensated resistances may be introduced. In this system we have everything compensated for. We must not have any coil that is going to vary with temperature at all. The coils that are to balance the change of resistance in the thermometer must be compensated. The box which I used was one that had been carefully compensated. There are only two of these in existence. One is owned by Professor Calendar of London and the other by myself. Professor Calendar made his and I made mine. I was over a year in making the box, and each coil was separately compensated and placed in the box; and there are a number of such coils so that it took a great deal of time.

39. The principle of the compensating lead was introduced by

Professor Calendar and renders the readings independent of the length of connecting wire. I am successfully operating a thermometer placed nearly a mile away from the place of observation.

Of course the details of construction were worked out carefully, such as placing the coils in lead tubing; everything worked fairly satisfactorily. A very sensitive galvanometer is required. For less accurate work the telephone can be used to replace the galvanometer.

40. Variations of resistance which determine the temperature are measured on a scale made in any convenient form. The battery is located as shown in the diagram, as is also the galvanometer circuit or telephone, which may be used in place of the galvanometer.

These scales can be graduated in any way that we wish and they may be made to read with any degree of accuracy. We can get a degree ten centimeters long or a degree one-tenth inch long. It does not make any difference. The bulb is connected with the wire, which is connected to a direct reading scale connected with a contact piece that slips along the wire. There is nothing to break the circuit for the use of the telephone except the interrupted contact.

41. The wire of a platinum coil is about 0.006 to 0.007 of an inch in diameter. The wire for the bridge may vary from sixteen thousandths to twenty-five thousandths, or even only ten thousandths of an inch. It depends on the condition.

42. Evaporation would have quite a considerable influence on the loss of heat and consequent formation of ice, but it is probable that it is small compared to the other agencies at work. At night we have a clear sky practically free from water vapor, and, therefore, the heat is almost unabsorbed. That is, it passes through the layers of air without any reflection back again. On a cloudy night the rays are reflected back into the water. General calculations can only be made under given conditions. The average heat we receive from the sun is about 0.05 calorie per square centimeter per second.

We know about what the surface of a black body will radiate or how much heat will be given off. Such surface radiation is about 0.0003 calorie per square centimeter per second per degree difference of temperature.

That seems very small, but if you consider what the radiation is from the large surface of a river it seems much greater.

43. Suppose we make a calculation of how much heat would be radiated at 32 degrees Fahr. to air of 0 degrees Fahr. We would find if we applied our calculations that about four and one-half pounds of ice would be formed per square yard per hour. Now if we consider the radiation from the water surface at night into space at absolute zero then we would get 80 pounds per square yard per hour formed. So you see how very much greater the radiation would be on a clear night and what a large proportion of ice would be formed. That is merely by radiation. Of course these calculations are based on assumptions which very seldom hold rigidly in actual conditions. From the river bottom there would be a smaller proportion of ice formed. We would get an inch of solid ice formed in about six or six and one-half hours.

44. The effect of pressure has a strong bearing on the question of the state of the ice, as it tends to lower the freezing point. One hundred and fifty atmospheres lowers the freezing point 1 degree Cent. One foot of water-pressure would lower the freezing temperature then about 0.0022 degree Cent., which would be about 0.0004 degree Fahr. This means that the undercooled surface layers if submerged deep enough would be at or above their freezing point. The ice crystals which on the surface would stick together from the fact that they were at temperatures below their freezing point would not stick together at a great depth.

45. Now the effect of falling water is to generate heat. One foot of fall would raise the temperature 0.00131 degree Fahr., or 0.0007 degree Cent. You see that water falling rapidly would have an important influence in counteracting the slight undercooling. But it would have to fall rapidly to counteract the cooling effect of air at or below the zero point. In the case of water flowing several feet in several hours this effect of heating by fall would be negligible. But if it was falling rapidly over say a head or a natural fall it would have an important influence. But it would be much smaller than calculation on account of the amount of mixed air; and the exact influences could only be determined by measurement.

DISCUSSION.

The President.—I am tempted to add a word myself on this paper because it is such a beautiful illustration of the value of the precise work of the physicist to the practicing engineer.

These temperature observations have been made by Professor Barnes with a degree of precision never before attained, ranging from the thousandth part of a degree Fahrenheit in some experiments, down to hardly more than a ten-thousandth part of a degree in others, and because of this extreme precision they have served to explain many of the phenomena of anchor ice with which the practicing engineer has to deal.

In my own practice, during the last winter, I have had occasion to make very careful studies of anchor ice on the St. Lawrence river at a great water power about a hundred miles upstream from Montreal, and nothing that I have found in scientific publications has been so illuminating and instructive as this work of Dr. Barnes in explaining many of the phenomena of anchor ice.

Those who have dealt much with water power in northern latitudes know that anchor ice is one of the greatest obstacles to keeping the wheels moving, and that there are many things about its formation and about the release of its grip on the bottom that have been very hard to explain. I recall a story told of an experience they once had at Lowell, Massachusetts, where, on attempting to hoist the gates that supplied the northern canal, the turbine by which the canal head gates were hoisted refused to work, and it was found to be completely choked with anchor ice. The factories' turbines could not be supplied and all was trouble and confusion while they slowly and laboriously worked the head gates up by hand. Finally, before this was accomplished, the sun rose to a point where it shone in the water in front of the turbine, and in a very few minutes the turbine started. It was suprising that that brief glimpse of the sunshine was sufficient to free the turbine from the anchor ice, but this work of Dr. Barnes, showing that the temperature of the water has to be raised only a few thousandths of a degree in order to cause the anchor ice to cease its grip, explains the matter.

In the St. Lawrence river, between Ogdensburg and Montreal, the anchor ice forms in great masses on the bottom of the

channel on cold nights at points where the current is so rapid that the surface of the river does not freeze over, but as soon as the sun comes out in the morning this anchor ice will let go its hold on the stones in the bed of the river and rise in broad sheets and over such large areas that the boatmen have learned to be very cautious about venturing out on the river near the rapids at such times; for these jelly-like masses may suddenly rise about the boat and impede rowing it or guiding it ashore. It has heretofore been very hard to understand how such a brief action of the sunshine could cause these masses of ice to let go their hold on the bottom. These experiments of Dr. Barnes show that the quantity of heat—the number of thermal units—necessary to raise the temperature of the water so that this ice will cease being "anchored" to the bottom is an exceedingly minute quantity.

Some Canadian engineers, and others interested in this subject, have recently made an application to the Carnegie Institution for a grant for the purpose of continuing these studies by Dr. Barnes, and we earnestly hope the wherewithal may be provided for this precise and accurate investigation. It promises much to the water power engineer.

Mr. William F. Mattes.—Mr. President, you touched upon one point that occurred to me as the paper was being read, and that is whether these experiments included any note of the difference between still water and water that is agitated. I remember an old experiment that we made when I was a boy, where, under certain conditions, a vessel of water could have its temperature lowered by the surrounding atmosphere to a little below the freezing point without giving any evidence of freezing, but that a little jar or shake would start crystals of ice shooting across its surface instantly. I thought possibly if that feature had not been taken into consideration it might modify the results of these experiments.

The President.—That was taken into consideration by Dr. Barnes, and my recollection is that he made certain experiments which showed that water absolutely quiet as, for example, contained in a vessel on the stone pier of a laboratory, could be cooled several degrees below 32 Fahrenheit without ice forming in it, but when thus undercooled, the simple act of sriking the vessel containing the water would cause ice crystals to form instantaneously through it. Where the water is in motion, as

it always is in a flowing canal or a river, there is always an amount of turmoil, eddying and motion going on which makes it impossible to cool the water more than a hundredth part of a degree below 32 Fahrenheit however intensely cold the atmosphere may be.

*Mr. Barnes.**—In reply to Mr. Mattes' question, it may be said that much attention has been devoted to the surfusion of water. As early as 1724 Fahrenheit found that a glass bulb filled with water and hermetically sealed could be cooled considerably below the freezing point without solidification setting in. Gay Lussac also, in 1836, cooled water under a layer of oil to -12 degrees Centigrade, but the slightest shake was found sufficient to cause ice to form. Dufour, in 1863, succeeded in cooling small drops of water suspended in another liquid of equal density to -20 degrees Centigrade. In all cases the water was as pure as possible and free from dust. Wherever water is kept perfectly quiet, and as free as possible from dissolved air, it is possible to supercool it. Agitation or the presence of the smallest ice crystal starts the solidification, after which supercooling is impossible. In view of these facts it is exceedingly improbable that running water with exposed surface, containing in nearly all cases quantities of denudated material and dissolved air, and subjected to wind agitation, can become cooled appreciably below the freezing point.

In some experiments which I carried out on the "Specific Heat of Supercooled Water" it was found very difficult to cool water with exposed surface more than 4 or 5 degrees below the freezing point without ice forming. The mere touching of the thermometer against the sides of the vessel under the surface of the water was enough to start the ice.

In regard to the natural growth of ice, it seems that Faraday had an idea that this took place even when the temperature of the surrounding air was considerably above the freezing point. He states that a small quantity of water surrounded on every side by ice has a *natural tendency to become ice*. This bears out the fact that two blocks of ice placed in contact do not unite unless they are moist. After trying to explain Faraday's conclusions in a number of ways, Professor J. Thomson finally pointed out that the analogy may be found in the crystallization of salts from their aqueous solutions. The matter is one, however, which

* Author's Closure, under the Rules.

requires further careful investigation, especially as to the rate of growth.

I am sure that it is only with an accurate knowledge of the physical laws underlying the formation of ice that we can hope to successfully handle the ice problem as it presents itself in its various aspects to the engineer.

If we cannot hope to cope with nature in preventing the formation of ice, we can certainly so guide our efforts as to minimize its influence on the great water power development of our country.

No. 1083.*

BELT CREEP.†

BY WM. W. BIRD, WORCESTER, MASS.
(Member of the Society.)

1. The question of the minimum amount of slip of a belt in transmitting power from one pulley to another, reduces itself to a question of creep, for it is possible to have belts large enough so that with proper tensions, there will be no regular slip. With a difference in tension on the two sides and elasticity in the belt, creep however is bound to take place. What does it amount to and what allowance should be made for it?

2. In Fig. 207 let A be the driver and B the driven, T_1 the tension in the tight side of the belt and T_2 in the slack side, the pulleys and belt running in the direction indicated. One inch of slack belt goes onto the pulley B at o and at or before the point p, it feels the effect of increased tension and stretches to $1 + s$ inches. It now travels from p to m and goes onto pulley A while stretched. At or before reaching the point n, as the tension decreases it contracts to one inch and so completes the cycle.

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* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussion on this general subject consult *Transactions* as follows:

No. 98, vol. 4, p. 165: "Belting to Connect Shafts which are not Parallel." J. B. Webb.

No. 202, vol. 7, p. 347: "Transmission of Power by Belting." (M. I. T. Experiments.)

No. 213, vol. 7, p. 549: "Experiments on the Transmission of Power by Belting." Wilfred Lewis.

No. 251, vol. 8, p. 529: "Comparative Value of Different Kinds of Leather Belting." S. Webber.

No. 568, vol. 15, p. 204: "Notes on Belting." Fred. W. Taylor.

No. 797, vol. 20, p. 136: "Variation of Belt Tension with Power Transmitted." Wm. S. Aldrich.

No. 808, vol. 20, p. 466: "Relation between the Initial Tension and Power Transmitted by a Belt." F. L. Emory.

3. With a light load the belt creeps ahead of the pulley B at or near the point p. If the load is heavy, the creep works towards the point o and the belt may slip; this also takes place when the belt tensions are too light even with small loads.

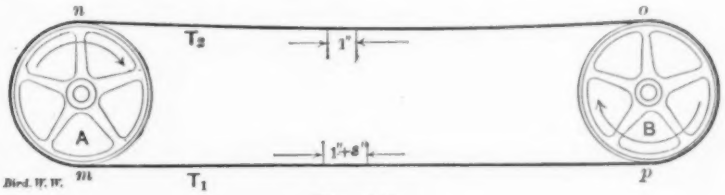

FIG. 207.

4. On the pulley A the belt creeps back relative to the pulley surface at n and as in B may extend to the point m when regular slip will occur.

Notation.

Let x = per cent. of slip.
v_1 = pitch velocity of driver.
v_2 = pitch velocity of driven.
t = difference in tension per square inch.
E = modulus of elasticity.
s = stretch in one inch due to t.

$$x = \frac{v_1 - v_2}{v_1}, \qquad \frac{v_1}{v_2} = \frac{1+s}{1}, \qquad \frac{E}{t} = \frac{1}{s}, \qquad x = \frac{s}{1+s} = \frac{t}{E+t}.$$

The following table gives the per cent. of creep for various tensions and values of the modulus of elasticity.

TABLE I.

PER CENT. CREEP.

Effective Tension per sq. in.	MODULUS OF ELASTICITY.			
	5,000	10,000	15,000	20,000
25	.49	.25	.17	.12
100	1.96	.99	.66	.50
125	2.44	1.23	.83	.62
150	2.91	1.48	.99	.74
175	3.38	1.72	1.15	.87

5. In order to determine a value for the modulus, a number of experiments were made on 4 and 6-inch single and double endless leather belts. The tension in the tight side was taken as 50 pounds per inch of width for single belts and 80 for double. The tension in the slack side, 20 for single and 32 for double. The belt testing machine in the Engineering Laboratory of the Worcester Polytechnic Institute was used, the dynamometer being removed so that the belts ran without load. The sum of the tensions being made twice the desired tension, first for the tight and then for the slack side, changing as rapidly as possible from one to the other, the belt running about one thousand feet per minute horizontally. Readings were taken on a scale which showed the distance between the centers of the two 18-inch pulleys which were used. In this way what might be called the running modulus of elasticity was determined for the given range of tensions. The following table gives the results, being the average of ten tests for each belt.

TABLE II.

Belt.	Area of Cross Section.	T_1	T_2	$T_1 - T_2$	Tension per Sq. In.	Length.	Stretch.	Modulus of Elasticity.	Corresponding per cent. of Creep.
4-in. single.	.86	200	80	120	139.53	295.3	2.095	19.665	.70
6-in. single.	1.34	300	120	180	134.33	293.6	2.025	19.475	.67
4-in. double	1.3	320	128	192	147.69	293.2	1.992	21.240	.67
6-in. double	2.02	480	192	288	142.57	294.0	1.962	21.360	.66

6. With an ordinary leather belt running under the conditions named, which are generally considered good practice, the creep should not be over one per cent.

Figs. 208 and 209 give the front and rear views of a model for showing creep. The two pulleys are the same size, one being connected to a dial and the other by gears and shafts to a pointer on the dial. The driver is turned by a crank and pinion while a brake on the driven supplies the load. In this way the creep is shown, and by using an elastic band the driver may run twice as fast as the driven without regular slip. The gears and pointer on the left hand side of the model are for the purpose of showing the speed of the belt as it goes onto each pulley, which is simply another way of showing creep.

7. In connection with these experiments it was found that the modulus of elasticity of leather belting was a very uncertain quan-

tity, the regular testing machine methods giving all kinds of results. That it varies in actual working condition has been shown by numerous experiments reported to this Society, which show that the sum of the tensions increases with an increase in the

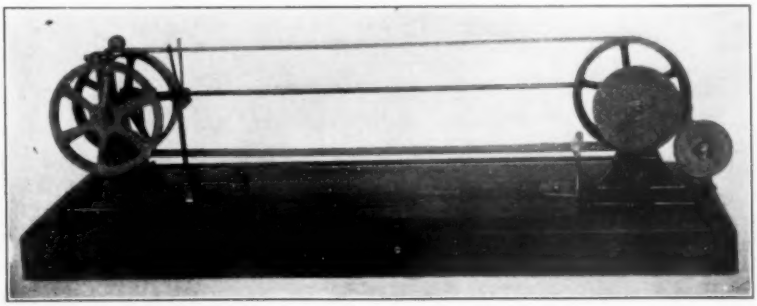

FIG. 208.

difference of tensions or load, the distance between centers remaining constant.

8. As the method employed in getting values of E in the tests in Table II gave uniform results, the same plan was used for an investigation as to how the modulus varied for smaller changes in the difference of tensions. These experiments were made first

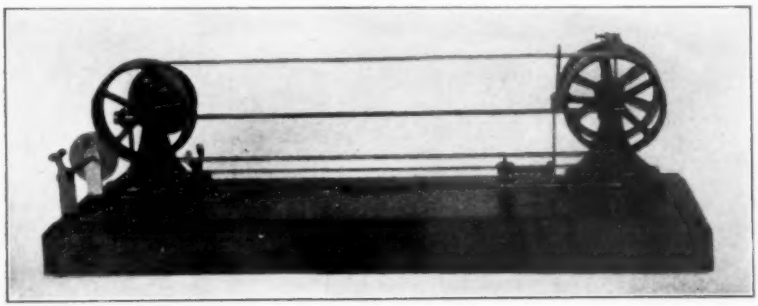

FIG. 209.

on a 4-inch single endless belt running about one thousand feet per minute without load. Starting with 300 pounds which would be about the sum of the tensions for ordinary conditions, it was decreased 25 pounds and then increased 25 and so on as given in Table III. This corresponds to what takes place in practice, the tension is uniform with no load, decreases in the slack side and increases in the tight as the load comes on. After allowing the

belt to run long enough at 300 pounds or 150 tension to adjust itself, the sums were changed as rapidly as possible and the distance between centers noted.

TABLE III.

No. Test.	Tension.	Length.	No. Test.	Tension.	Length.
1	150.0	296.60	9	200.0	297.32
2	137.5	296.40	10	87.5	295.38
3	162.5	296.78	11	212.5	297.48
4	125.0	296.18	12	75.0	295.08
5	175.0	296.96	13	225.0	297.64
6	112.5	295.94	14	62.5	294.76
7	187.5	297.14	15	237.5	297.78
8	100.0	295.68			

Fig. 210 shows the curve for these results.

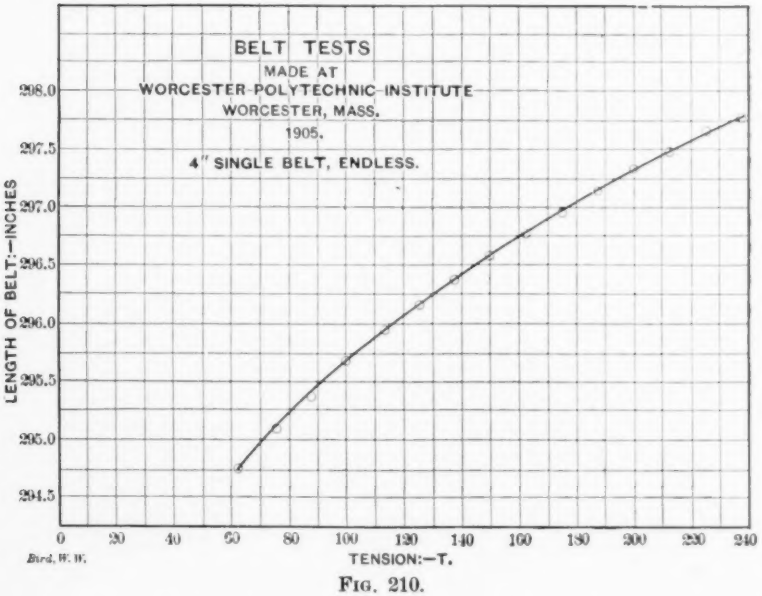

FIG. 210.

9. By combining these experimental results so that the sum of the tensions in each case is 300, the lengths of the belt for various differences of tension are obtained by taking half of the sum of the lengths corresponding to the given tensions.

10. Table IV. gives the result of these combinations and the lower line in Fig. 211, the corresponding curve.

TABLE IV.

Tests.	$T_1 + T_2$	$T_1 - T_2$	Length.
1	300	0	296.60
2–3	300	25	296.59
4–5	300	50	296.57
6–7	300	75	296.54
8–9	300	100	296.50
10–11	300	125	296.43
12–13	300	150	296.36
14–15	300	175	296.27

TABLE V.

Tests.	$T_1 + T_2$	$T_1 - T_2$	Length.
201	400	103 8	297.22
202	400	138.4	297.14
203	400	173.0	297.02
204	400	207.0	296.88

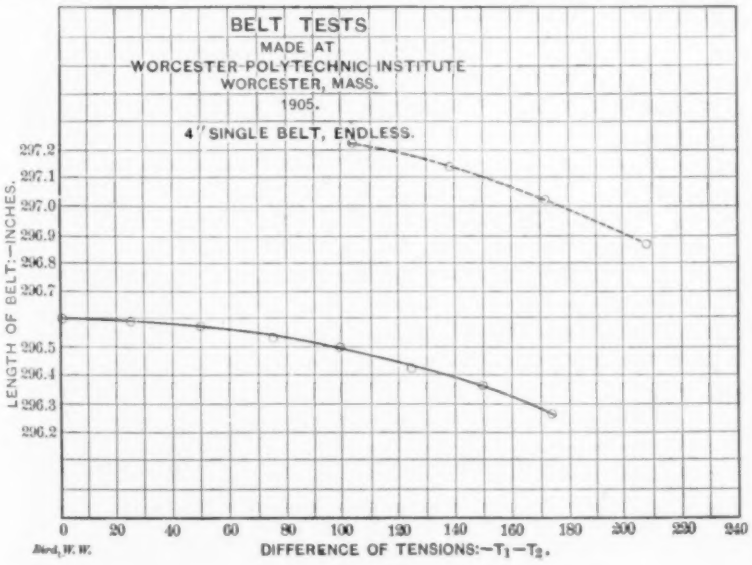

FIG. 211.

11. Regular tests of the same belt run on the belt testing machine before the absorption dynamometer was taken off, gave the results given in Table V. The sum of the tensions, however, was greater in these tests, but the dotted curve in Fig. 211 shows that the belt decreases in length with increase of load in actual tests

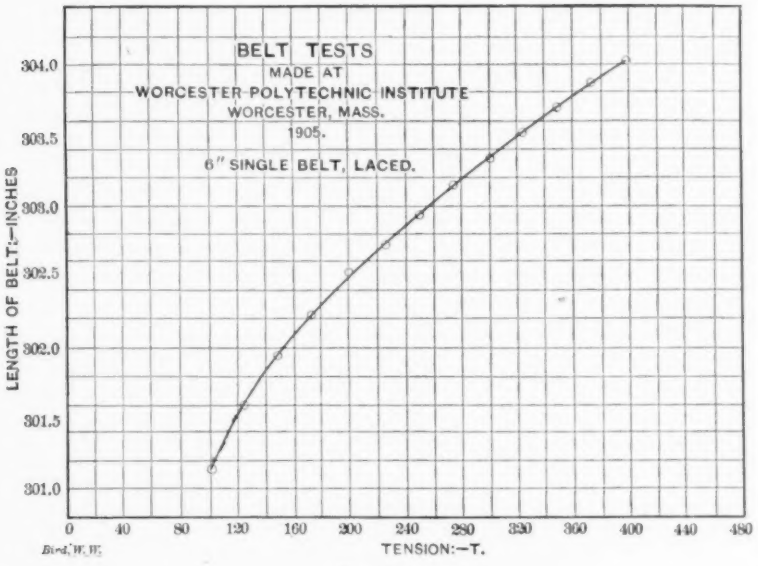

FIG. 212.

in about the same way as shown by the combination curve in the same figure.

12. In actual practice where the distance between the centers remains constant, the sum of the tensions increases with the load for the reason that starting with a uniform tension, as the load comes on, the amount of stretch on the tight side, corresponding to the increased tension is not as much as is required to relieve the tension in the slack side an equal amount. Therefore in order to get a given difference in tension, it is necessary to increase the tight side more than the slack side is decreased, or in other words the sum of the tensions increases with the load.

13. Table VI. gives the results of tests on a 6-inch single laced

TABLE VI.

Test.	Tension.	Length.	Test.	Tension.	Length.
1	250	302.92	8	150	301.94
2	225	302.72	9	350	303.68
3	275	303.12	10	125	301.58
4	200	302.50	11	375	303.86
5	300	303.32	12	100	301.14
6	175	302.22	13	400	304.02
7	325	303.50			

TABLE VII.

Tests.	$T_1 + T_2$	$T_1 - T_2$	Length.
1.....	500	0	302.92
2–3.....	500	50	302.92
4–5.....	500	100	302.91
6–7.....	500	150	302.86
8–9.....	500	200	302.81
10–11.....	500	250	302.72
12–13	500	300	302.58

belt, what would be considered as an old belt, and Fig. 212 gives the corresponding curve. Table VII. is the combination of the tests and Fig. 213 the curve.

14. The modulus of elasticity varies in these tests from 12,000 pounds per square inch to about 30,000. The increase in this is largely due to the time element, for the belt would stretch more if it had time, but before an inch of the belt, when it is running, can stretch to a natural length for T_1, it returns to T_2 and starts to contract. The conclusion drawn is that the modulus increases with the tension when the belt is running and is probably larger the greater the speed.

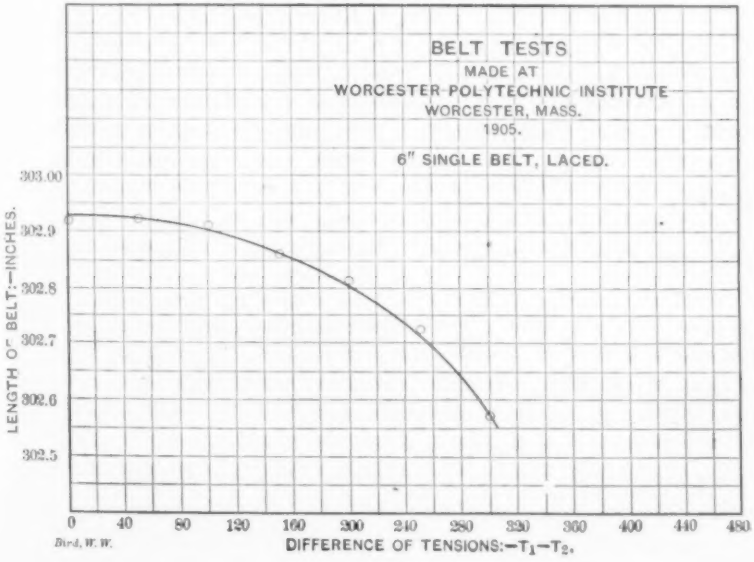

FIG. 213.

15. Allowing the belt to adjust itself for each tension statically the average value of E for these belts was about 12,000, while 20,000 is a fair value for ordinary working conditions.

16. Of course the sag in the belt varies with the tensions and has some effect on the question of the length of the belt. In these tests the 4-inch belt showed a difference of less than half an inch in the sag for the range of tensions used and still less in the 6-inch belt. As the pulleys were about 10 feet apart it can be seen that the error due to this is small.

17. In conclusion, the answer to the opening question is that for the common leather belt running under ordinary conditions the creep should not exceed one per cent. While this is sometimes called legitimate slip, it is an actual loss of power and can not be avoided by belt-tighteners or patent pulley coverings.

DISCUSSION.

Mr. Geo. H. Barrus.—Mr. Bird's interesting paper refers to experiments on belts connecting two pulleys eighteen inches in diameter, and as a result of these experiments he states that for 4-inch and 6-inch single and double leather belts the amount of creep should not be over one per cent.

It would be interesting to know whether the author has made observations on other sizes of pulleys than the ones mentioned, and on belts connecting pulleys of different diameters. If he has not, would he state his opinion as to what effect a change of diameter either of one pulley or both would have on the conclusion given.

The formula in paragraph 4 refers to the "pitch velocity" of the two pulleys. Does he mean by this the velocity of the outside circle of the rim of the pulleys, or does he refer to the velocity of some point in the belt corresponding to a circle of larger diameter than that of the rim? Also in the case of a crowned pulley, would the "pitch velocity" be that of the edges of the rim or that of the highest point of the crown?

It would be interesting, also, if the author of the paper would state the precise method he would follow in calculating the theoretical speed of a driven pulley when the two pulleys are of unequal diameters and when both are crowning, that is, the speed of the driven pulley assuming an entire absence of creep.

*Prof. W. W. Bird.**—In reply to Mr. Barrus, I would say that our experiments on belt creep were all made on the eighteen-inch pulleys. Creep, in my opinion, is, however, independent of the diameters of the pulleys, other things being equal, and depends upon the difference in the tensions of the two sides and on the modulus of elasticity of the belt. The element of time probably makes some difference in the creep, the slower the belt is moving the more time a unit length has to stretch or contract as the tene sion changes. The distance between centers or the length of the belt and the diameters of the pulleys would be factors in this time-element for adjustment, and so may have a slight effect upon the creep, on this account.

The pitch velocity means the velocity of the middle point of the belt on a cross-section at the mean diameter of the pulley. Experiments have shown that this assumption is fairly correct for pulleys with an ordinary crown. To assume an entire absence of creep, as Mr. Barrus suggests, gives us a problem entirely aside from the purpose of this paper. If power is to be transmitted and the belt has elasticity, there is bound to be creep.

* Author's Closure, under the Rules.

No. 1084.*

THE TRANSFER OF HEAT AT HIGH TEMPERATURES.

BY FRANK C. WAGNER, TERRE HAUTE, IND.

(Member of the Society.)

1. The experiments referred to in this paper were made several years ago for commercial purposes, and for this reason the methods adopted were perhaps not those best suited to secure scientific results. It is believed, however, in view of the meager data at present available concerning the transfer of heat at high temperatures, that what follows will be of sufficient interest to justify the presentation of this paper.

2. The primary object of the experiments was to determine the time required to raise plates of iron and steel to a welding temperature in an open hearth regenerative furnace. Some preliminary experiments were made by heating plates in a muffle of the size commonly used in chemical laboratories.

3. Four different samples of metal were experimented upon. These samples were cut from stock plates used in the manufacture of butt-weld pipe. The dimensions of the samples were as follows:

No. of Plate.	Material.	Length.	Width.	Thickness.
1.....	Iron.	6 inches.	1.25 inches.	.104 inch.
2.....	Iron.	6 "	1.25 "	.117 "
3.....	Steel.	6 "	1.25 "	.111 "
4.....	Iron.	6 "	1.25 "	.129 "

4. The temperature of the specimen plate was measured by a Le Chatelier thermocouple. A small hole about a thirty-second of an inch in diameter was drilled edgewise through the plate near the center of its length. The wire of the thermocouple was threaded through this hole and the junction of the two wires of

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

the couple was located at about the center of the plate. Strips of mica were wedged about the wires so as to insure that the wires did not make contact with the iron plate near the edges. A second thermocouple was located in the furnace adjacent to the plate for the purpose of measuring the furnace temperatures.

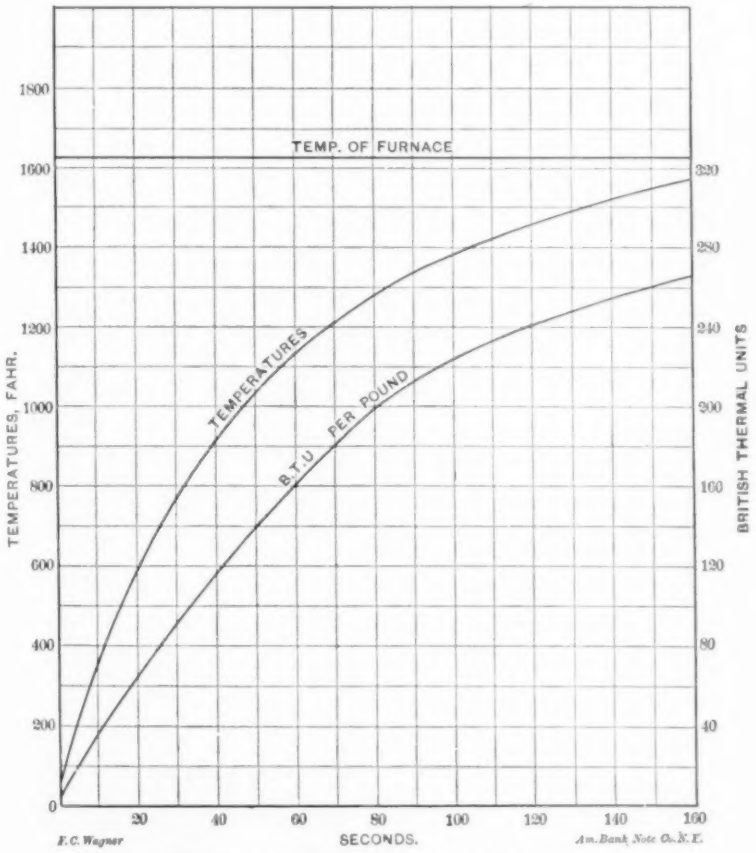

FIG. 214.—PLATE 1 IN MUFFLE. TEST 1.

5. The experiment was made in the following manner. The furnace was brought up to the desired temperature and its temperature measured. The plate with the inserted thermocouple was then quickly placed in the furnace and the opening closed. The exact time of placing the plate in the furnace was noted by an observer, who thereafter called out each five-second interval to another ob-

server at the galvanometer. In this way the temperature of the plate was determined at five-second intervals until the plate reached substantially the temperature of the furnace. At the end of the experiment the temperature of the furnace itself was again read and the mean of the temperatures before and after was taken as

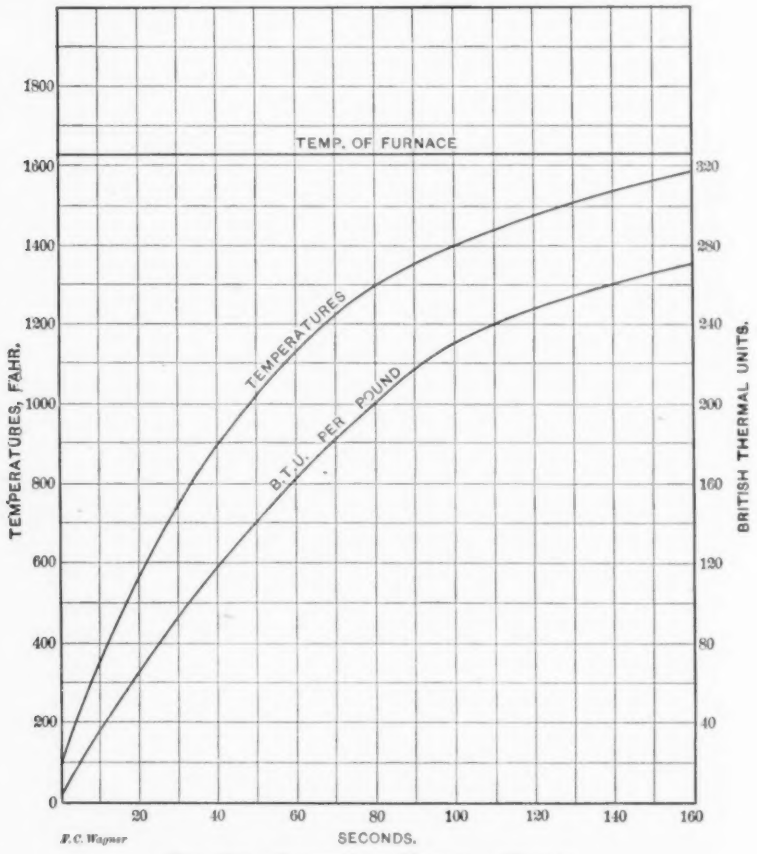

FIG. 215.—PLATE 1 IN MUFFLE. TEST 2.

that during the test. The results obtained are shown in Figs. 214 to 221.

6. In all of these tests the transfer of heat was principally by radiation. When the muffle was used the damper in its vent pipe was closed as was also the opening in front. In the open hearth furnace the flames passed across the top of the furnace without

coming into contact with the floor of the furnace. The openings for inserting material were kept closed by sheet iron dampers during the tests.

7. It is possible from these data to determine the rate of transfer of heat from the furnace to the iron. The specific heat of iron

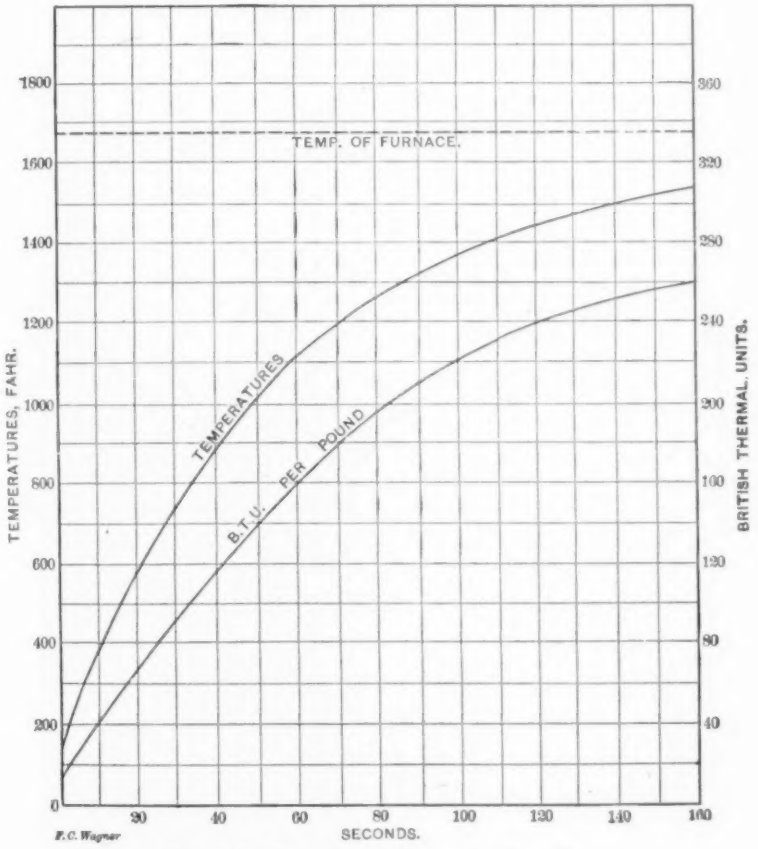

FIG. 216.—PLATE 2 IN MUFFLE. TEST 1.

varies with the temperature. Pionchon in *Comptes Rendus*, Vol. 103, page 1123, gives it as follows:

From 0° to 660° Cent., Specific Heat

$$= 0.11012 + 0.000050666t + 0.000000163998t^2.$$

From 660° to 720° Cent., Specific Heat

$$= 0.57803 - 0.00287196t + 0.000003585t^2.$$

From 720° to 1000° Cent., Specific Heat = 0.218.
From 1050° to 1200° Cent., Specific Heat = 0.19887.

In the formulas, t is the temperature in degrees Centigrade for which the formula gives the specific heat. By using these values a curve was constructed showing the heat units above 32 degrees

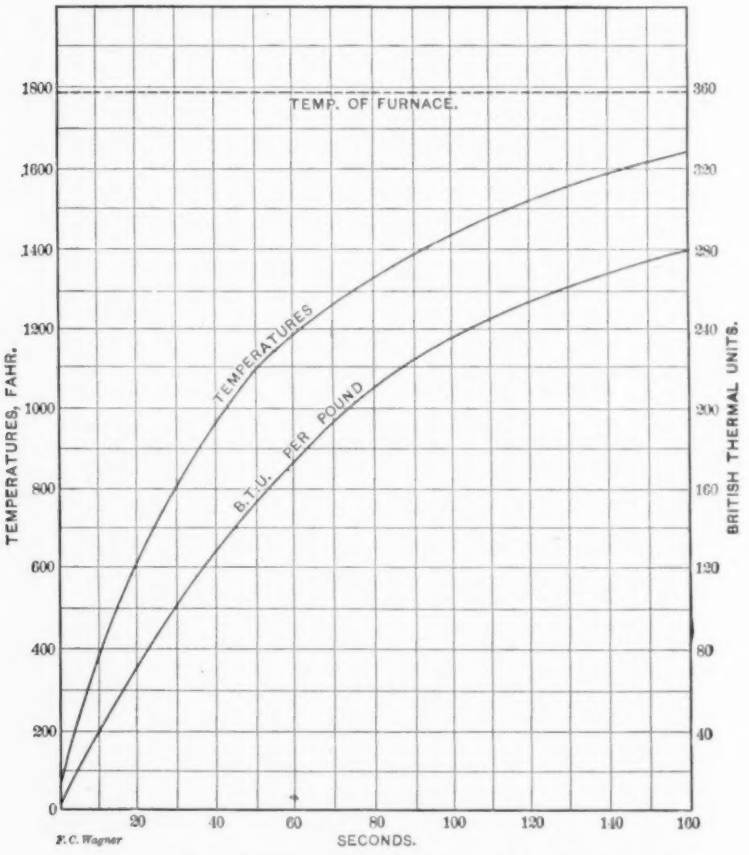

FIG. 217.—PLATE 2 IN MUFFLE. TEST 2.

Fahr. in a pound of iron at any temperature. By this means the curves in Figs. 214 to 221 labelled British Thermal Units per Pound were constructed. From these latter curves the slope of the tangent at any temperature gives the heat units absorbed per pound per second.

8. In the calculations it has been assumed that a plate laid on

the bottom of the muffle or furnace receives heat through both the top and the bottom surfaces from a source at the temperature of the furnace. This is strictly true only at the beginning of the heating period. The bottom of the muffle was covered with bone ash and it is quite probable, owing to its low heat conductivity,

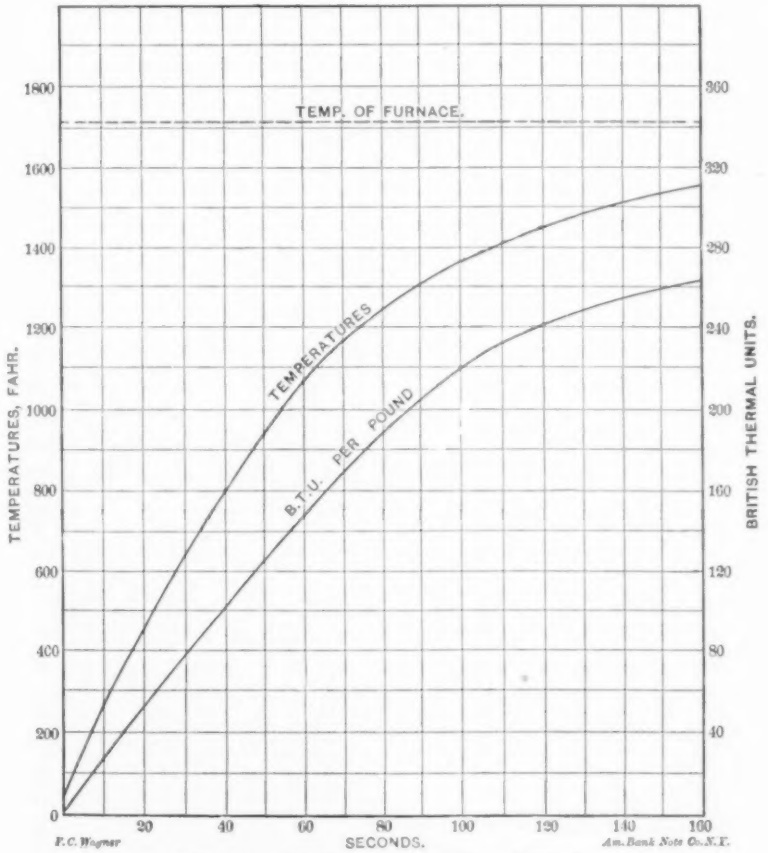

FIG. 218.—PLATE 3 IN MUFFLE. TEST 1.

that less heat was received by the plate through its lower surface than through its upper. In the case of the open hearth furnace the plate was placed upon a fire brick, whose capacity for storing heat was considerable, and whose heat conductivity was also sufficient to maintain a high temperature at the surface exposed to the iron plate. It seems probable that in this case the

heats received through the top and bottom surfaces respectively were nearly equal to each other.

9. It has also been assumed that the plate was at substantially the same temperature in the interior of the metal where the

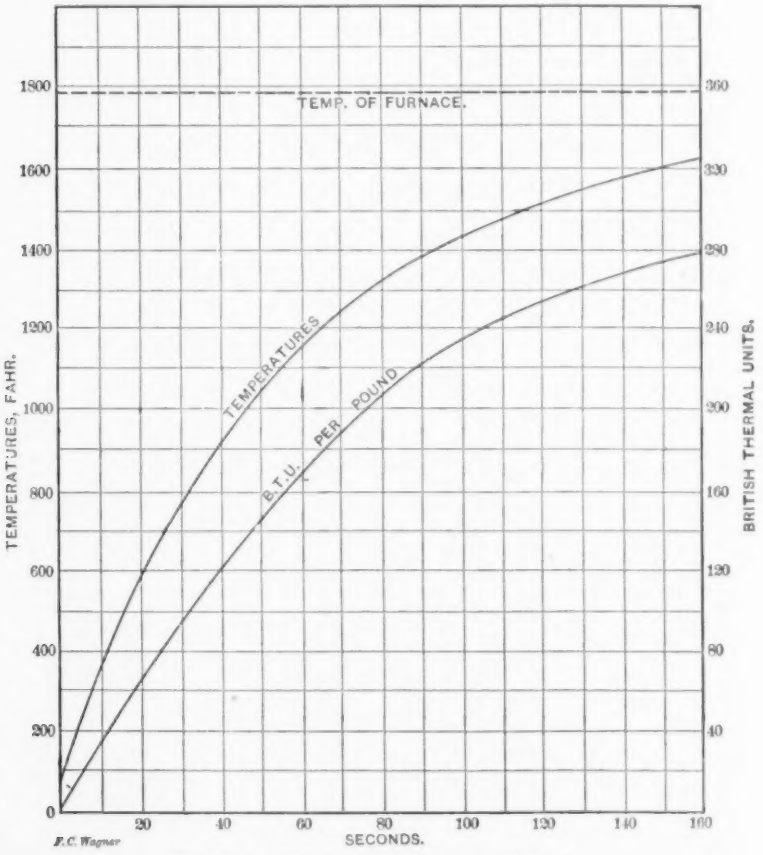

FIG. 219.—PLATE 3 IN MUFFLE. TEST 2.

thermocouple was located as at the surface. That this is practically true is evident from the following.

10. The heat-conductivity of iron is 0.16 small calories per second per square centimeter for one centimeter of thickness and one degree centigrade difference of temperature, as found by Forbes. This corresponds to 0.0009 British thermal units per square inch for one inch thickness and one degree Fahrenheit dif-

ference of temperature. The thickest plate experimented with was 0.129 inch thick, so that the heat was conducted 0.065 inch in passing from the surface to the thermocouple. A difference of temperature of one degree Fahrenheit would consequently transmit 0.0138 heat units per square inch per second from the surface to the center.

11. The maximum rate at which heat was absorbed in the muffle experiments was 0.077 heat units per square inch per second. This would require that the surface of the plate be 5.6 degrees Fahr. hotter than the center. Since the temperature of the plate was at the same time rising at the rate of 35 degrees per second, the lag in temperature of the thermocouple behind the absorbing surface was less than one-sixth of a second.

12. In the open hearth furnace experiments the maximum rate of heat absorption was 0.26 heat unit per square inch per second, which requires a difference in temperature of 19 degrees Fahr. At the same time the temperature was rising at the rate of 90 degrees per second, showing a probable lag of about one-fifth of a second.

13. Moreover, since in all the calculations only the rate of temperature rise has been used, it is the *difference* of the lags at the beginning and end of a five-second interval that introduces an error, and this difference is certainly less than the errors of observation.

14. The heat absorbed per square inch of surface per second has been calculated from the curves of British thermal units per pound by multiplying the B. t. u. per pound per second by the weight of a square inch of the plate experimented upon and dividing by two, upon the assumption that both surfaces are active in absorbing heat. The results obtained are given in Tables I. to V.

15. A comparison of these results with Stefan's law of the radiation of heat shows a very satisfactory agreement. According to Stefan the heat absorbed or radiated is proportional to the difference of the fourth powers of the absolute temperatures. Expressed algebraically, $Q = \sigma (\tau_1^4 - \tau_2^4)$, where τ_1 is the absolute temperature of the furnace and τ_2 is the absolute temperature of the plate being heated.

16. Tables I. to V. give the values of the constant σ when Q is expressed in British thermal units per square inch per second and τ_1, τ_2 are in degrees absolute on the Fahrenheit scale.

TABLE I.

Plate 1 in Muffle.

Temperature of Plate.		Temperature of Furnace.		British Thermal Units Absorbed per Sq. Inch per Second.		Values of $\sigma \times 10^{12}$.	
Degrees Fahr.	Degrees Absolute.	Degrees Absolute. Test 1.	Degrees Absolute. Test 2.	Test 1.	Test 2.	Test 1.	Test 2.
239	700	2,073	2,146	.0491	.0491	2.69	2.34
339	800	"	"	.0434	.0434	2.40	2.08
439	900	"	"	.0434	.0419	2.43	2.04
539	1,000	"	"	.0434	.0405	2.48	2.00
639	1,100	"	"	.0419	.0405	2.46	2.05
739	1,200	"	"	.0390	.0390	2.38	2.04
839	1,300	"	"	.0361	.0376	2.31	2.05
939	1,400	"	"	.0347	.0347	2.37	2.00
1,039	1,500	"	"	.0347	.0318	2.59	1.97
1,139	1,600	"	"	.0292	.0318	2.45	2.17
1,239	1,700	"	"	.....	.0289	....	2.25
1,339	1,800	"	"	.0173	.0207	2.18	1.94
1,439	1,900	"	"	.0116	.0144	2.13	1.77
		Average				2.41	2.06

TABLE II.

Plate 2 in Muffle.

Temperature of Plate.		Temperature of Furnace.		British Thermal Units Absorbed per Sq. Inch per Second.		Values of $\sigma \times 10^{12}$.	
Degrees Fahr.	Degrees Absolute.	Degrees Absolute. Test 1.	Degrees Absolute. Test 2.	Test 1.	Test 2.	Test 1.	Test 2.
239	700	2,136	2,246	.....	.0651	....	2.93
339	800	"	"	.....	.0602	...	2.73
439	900	"	"	.0439	.0553	2.18	2.54
539	1,000	"	"	.0423	.0520	2.14	2.42
639	1,100	"	"	.0406	.0465	2.10	2.21
739	1,200	"	"	.0390	.0446	2.08	2.19
830	1,300	"	"	.0390	.0423	2.18	2.16
939	1,400	"	"	.0374	.0400	2.21	2.15
1,039	1,500	"	"	.0342	.0390	2.17	2.24
1,139	1,600	"	"	.0325	.0358	2.28	2.25
1,239	1,700	"	"	.0260	.0342	2.09	2.42
1,339	1,800	"	"	.0195	.0293	1.90	2.45
1,439	1,900	"	"	.0130	.0163	1.67	1.73
1,539	2,000	"	"	.0065	.0120	1.36	1.86
1,639	2,100	"	"	.....	.0059	....	1.95
		Average				2.03	2.28

TABLE III.

Plate 3 in Muffle.

Temperature of Plate.		Temperature of Furnace.		British Thermal Units Absorbed per Sq. Inch per Second.		Values of $\sigma \times 10^{12}$.	
Degrees Fahr.	Degrees Absolute.	Degrees Absolute. Test 1.	Degrees Absolute. Test 2.	Test 1.	Test 2.	Test 1.	Test 2.
139	600	2,178	2,241	.0401	.....	1.79	....
239	700	"	"	.0370	.0463	1.66	1.85
339	800	"	"	.0370	.0463	1.68	1.86
439	900	"	"	.0370	.0432	1.70	1.76
539	1,000	"	"	.0370	.0432	1.72	1.78
639	1,100	"	"	.0370	.0401	1.76	1.69
739	1,200	"	"	.0370	.0401	1.81	1.73
839	1,300	"	"	.0370	.0386	1.88	1.72
939	1,400	"	"	.0370	.0370	1.98	1.73
1,039	1,500	"	"	.0355	.0370	2.03	1.83
1,139	1,600	"	"	.0309	.0339	1.93	1.82
1,239	1,700	"	"	.....	.0309	....	1.83
1,339	1,800	"	"	.0200	.0247	1.67	1.67
1,439	1,900	"	"	.0123	.0154	1.30	1.26
1,539	2,000	"	"	.0093	.0117	1.42	1.27
1,639	2,100	"	"	.0062	.0077	2.02	1.33
		Average.....				1.76	1.67

TABLE IV.

Plate 4 in Muffle.

Temperature of Plate.		Temperature of Furnace.	British Thermal Units Absorbed per Sq. Inch per Second.	Values of $\sigma \times 10^{12}$.
Degrees Fahr.	Degrees Absolute.	Degrees Absolute.		
239	700	2,171	.077	3.58
339	800	"	.0716	3.29
439	900	"	.0645	3.00
539	1,000	"	.0573	2.71
639	1,100	"	.0430	2.07
739	1,200	"	.0376	1.87
839	1,300	"	.0366	1.89
939	1,400	"	.0348	1.89
1,039	1,500	"	.0334	1.94
1,139	1,600	"	.0323	2.06
1,239	1,700	"	.0305	2.20
1,339	1,800	"	.0197	1.68
1,439	1,900	"	.0126	1.37
		Average.....		2.27

TABLE V.

PLATE 4 IN OPEN HEARTH FURNACE.

TEMPERATURE OF PLATE.		TEMPERATURE OF FURNACE.		BRITISH THERMAL UNITS ABSORBED PER SQ. INCH PER SECOND.		VALUES OF $\sigma \times 10^{15}$.	
Degrees Fahr.	Degrees Absolute.	Degrees Absolute. Test 1.	Degrees Absolute. Test 2.	Test 1.	Test 2.	Test 1.	Test 2.
139	600	2,836	2,871	.2105	.2636	3.26	3.89
339	800	"	"	.2105	.2636	3.28	3.91
539	1,000	"	"	.2105	.2636	3.31	3.94
739	1,200	"	"	.2105	.2636	3.37	4.00
939	1,400	"	"	.2105	.2636	3.47	4.11
1,139	1,600	"	"	.2105	.2636	3.62	4.29
1,239	1,700	"	"	.2105	.2636	3.74	4.43
1,339	1,800	"	"	.2008	.2636	3.71	4.35
1,439	1,900	"	"	.1865	.2295	3.61	4.18
1,539	2,000	"	"	.1721	.2080	3.53	4.01
1,639	2,100	"	"	.1578	.1721	3.49	3.55
1,739	2,200	"	"	.1363	.1434	3.30	3.22
1,839	2,300	"	"	.1147	.1291	3.13	3.23
1,939	2,400	"	"	.0968	.1112	3.07	3.20
2,039	2,500	"	"	.0896	.1004	3.50	3.48
2,139	2,600	"	"	.....	.0717	....	3.23
2,239	2,700	"	"	.....	.0430	....	2.92
2,339	2,800	"	"	.....	.0210	....	3.34
		Average.....				3.43	3.74

17. The best experimental value for the coefficient of emissivity at high temperatures that I have been able to find is that obtained by Kurlbaum and quoted in Preston's Theory of Heat. The value of σ there given is 5.32×10^{-12} watts per square centimeter per unit value of the function $\tau_1^4 - \tau_2^4$, the values of τ_1 and τ_2 being in Centigrade degrees absolute. Reduced to the units used in this paper σ becomes 3.081×10^{-15} British thermal units per square inch per unit function $\tau_1^4 - \tau_2^4$ expressed in Fahrenheit units absolute. This value lies between the extreme values calculated in this paper.

18. The values of σ found above do not show systematic variations within a single test, and the agreement among the different tests is fairly satisfactory.

19. The average value of $\sigma \times 10^{15}$ was for plate 1, 2.23; for plate 2, 2.15; for plate 3, 1.72; for plate 4 in muffle, 2.27. Plates 1, 2 and 4 were of wrought iron, while plate 3 was of steel. The average then for all the tests made upon wrought iron plates in the muffle was 2.22. It appears that the coefficient of emissivity for steel is less than for wrought iron by about twenty per cent.

20. As between tests 1 and 2 upon plate 4 in the open hearth furnace, it is necessary to explain that toward the close of test 1 the flux ran down upon the wires of the thermocouple, making contact between the wires and the iron plate at the edges of the plate.

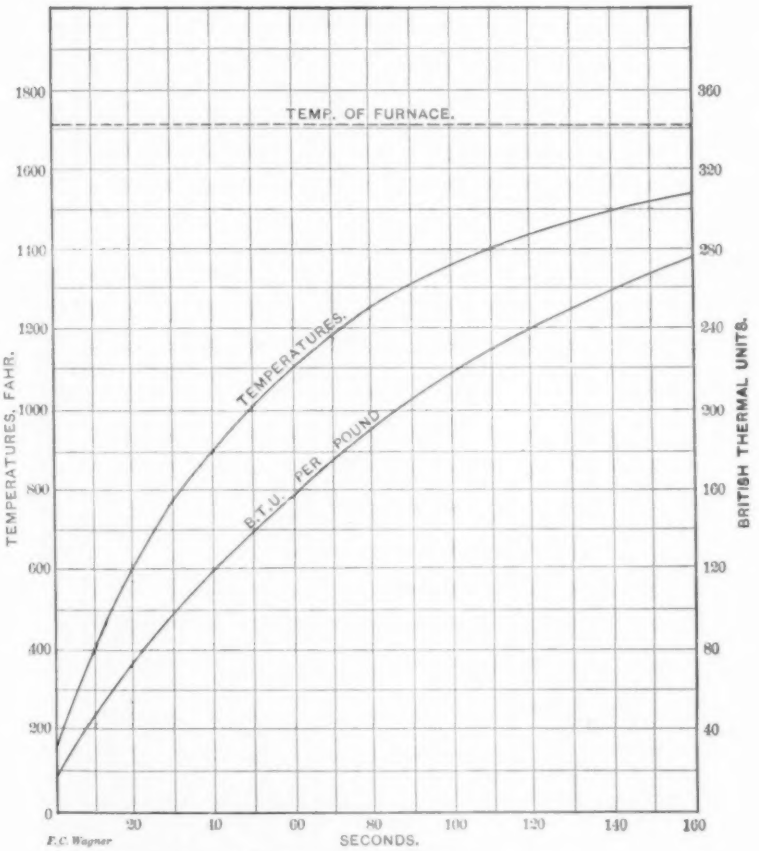

FIG. 220.—PLATE 4 IN MUFFLE. TEST 1.

When the second test was made, therefore, the thermocouple measured the temperature at the edges of the plate. Evidently the temperature at the edge rises faster than at the middle of the plate, because more surface is exposed to the radiant heat. This is very clearly shown in Fig. 221, where both tests have been plotted on the same diagram. The place where the curves intersect shows

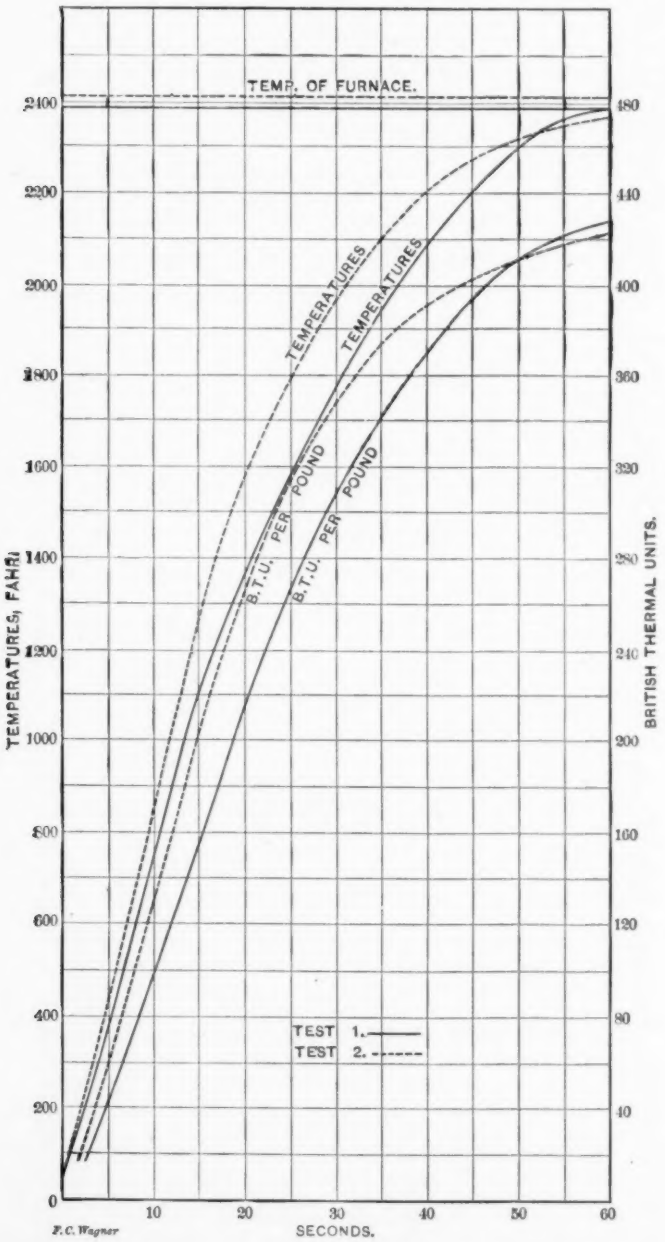

FIG. 221.—PLATE 4 IN OPEN HEARTH FURNACE. TESTS 1 AND 2.

approximately when the flux made contact with the wires of the thermocouple.

21. One striking result of these tests is brought out by comparing the times required to bring the plate up to the temperature of the furnace. In the muffle tests about three minutes were required, while in the open hearth tests one minute sufficed to heat the plate substantially to the furnace temperature.

22. If the law of Stefan holds for the radiation and absorption of heat under practical working conditions, as seems to be the fact, then some very significant relationships exist in the transfer of heat to the heating surface of a steam boiler. Compare two cases where the furnace temperatures are 2,000 degrees and 2,500 degrees Fahr. respectively, and a portion of the heating surface is exposed to direct radiation from the fire. The heats transferred by radiation will be as 37 to 77 in the two cases. In other words, adding 500 degrees to the temperature of the fire more than doubles the radiant heat transmitted to the water in the boiler.

No. 1085.*

PERFORMANCE OF A SUPERHEATER.

BY A. BEMENT, CHICAGO, ILL.
(Member of the Society.)

1. The apparatus consisted of a Babcock & Wilcox boiler of approximately 5,000 square feet of heating surface, of a height of fourteen tubes, and a Babcock & Wilcox superheater of approximately 1,000 square feet of heating surface, served by a chain grate stoker with a grate area of 75 square feet, which discharged the gases of the fire from under an ordinary short ignition arch directly among the tubes of the boiler.

2. The diagrams shown are plotted from a large number of experiments. The great variation in capacity produced was due entirely to the size of the coal used by the stoker, and the amount of ash it contained. In calculating the work developed in the superheater, the specific heat of superheated steam was taken as 0.75, although a higher figure is probably nearer correct, as the steam pressure was 180 pounds.

3. Fig. 222 shows relation of degree of superheat to total horse-power developed. It is apparent that the capacity increases at a much greater ratio than does the temperature of the steam, or, in other words, that the temperature tends in a measure to remain constant. For example, for ordinary ranges of capacity required in service, which may be taken as from 500 to 800 horse-power, the temperature of superheat ranges from 145 to 182 degrees, indicating that this combination of boiler and superheater probably produces as uniform a degree of superheat as would be obtained from a direct fired superheater in usual service. Throughout the whole range of the experiments, the total horse-power varied by approximately 850 per cent., while the degree of superheat varied by approximately 100 per cent. The condition of

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

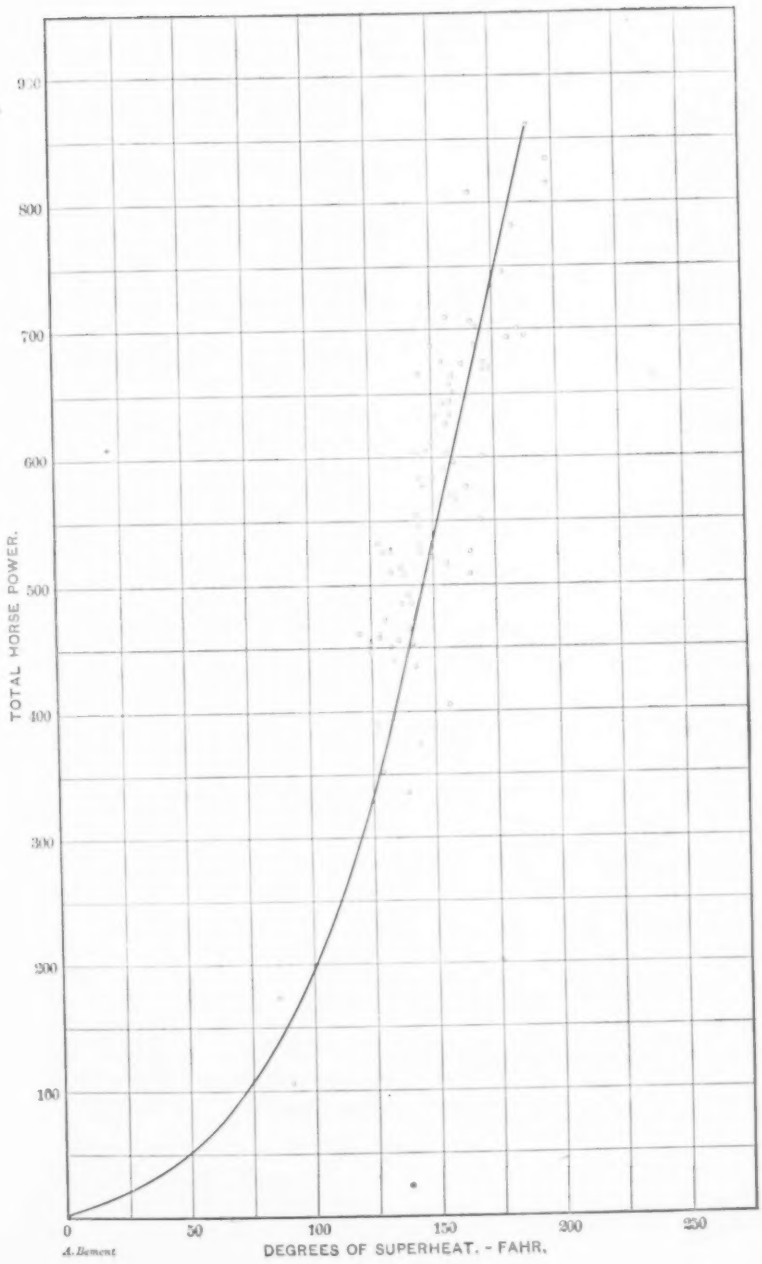
900
800
700
600
500
400
300
200
100
TOTAL HORSE POWER.
0
50
100
150
200
250
DEGREES OF SUPERHEAT. - FAHR.
A. Bement

FIG. 222.

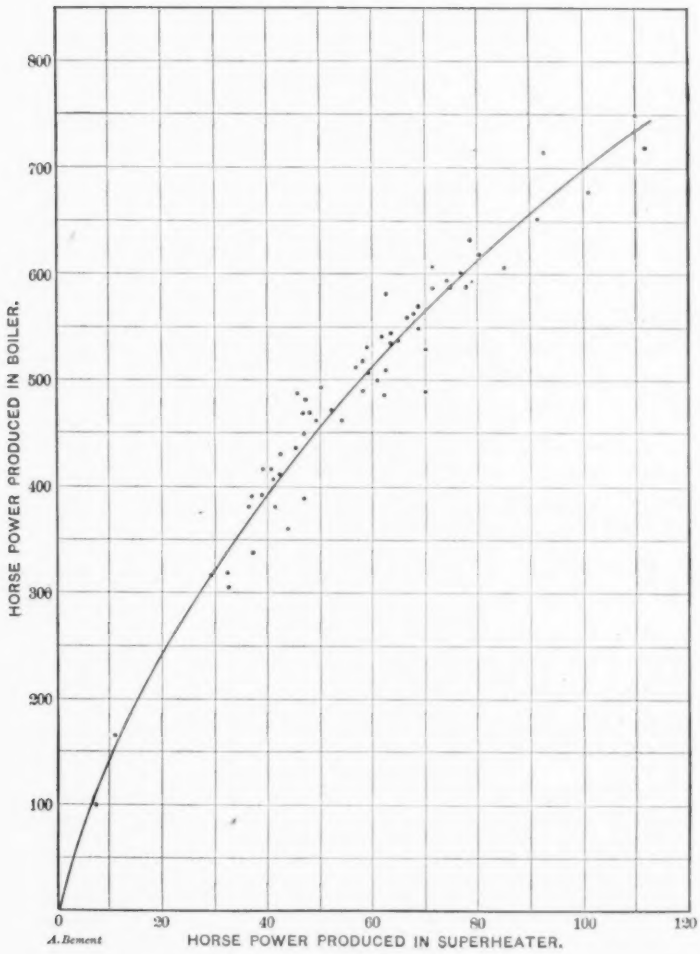
HORSE POWER PRODUCED IN BOILER.
800
700
600
500
400
300
200
100
0
20
40
60
80
100
120
A. Bement
HORSE POWER PRODUCED IN SUPERHEATER.

FIG. 223.

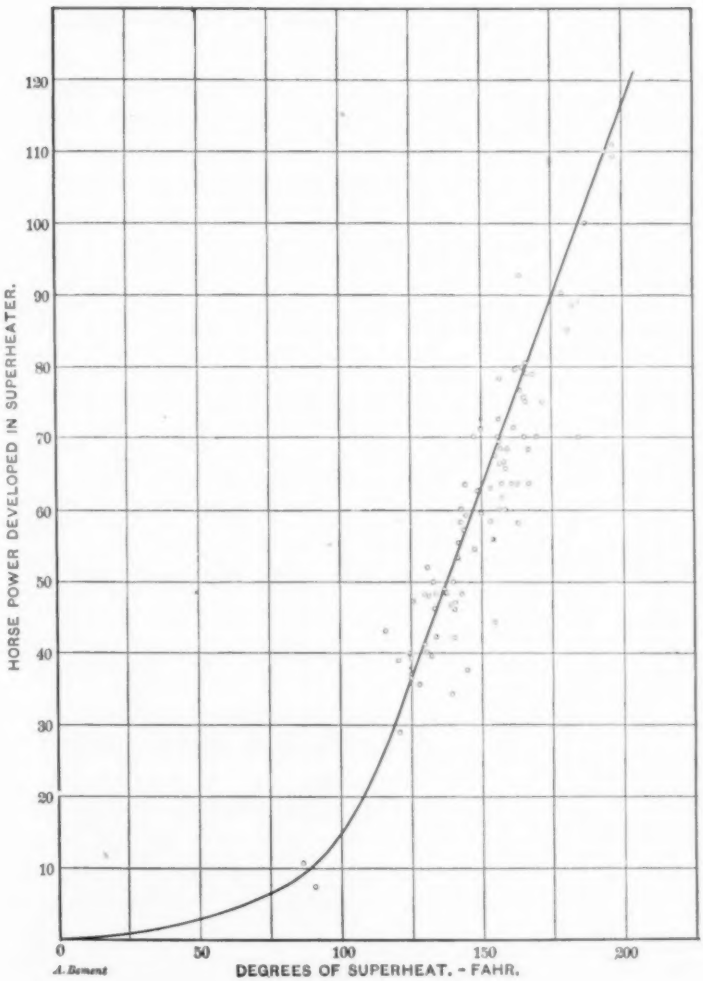
HORSE POWER DEVELOPED IN SUPERHEATER.
120
110
100
90
80
70
60
50
40
30
20
10
0
50
100
150
200
DEGREES OF SUPERHEAT. - FAHR.
A. Bement

FIG. 224.

combustion had much to do with the formation of this curve, because the large capacities were produced by the best coal, or, in other words, the coals of best size and greatest freedom from ash. It was, in fact, such fuels that made the large capacities possible, the largest horse-power being produced by coals specially selected, by screening out of a large quantity the most favorable sizes. For this reason, the condition of combustion was better as the horse-power increased, which produced a higher initial temperature, causing more heat to flow into the boiler and less to the superheater. Therefore, the tendency was for work done in the

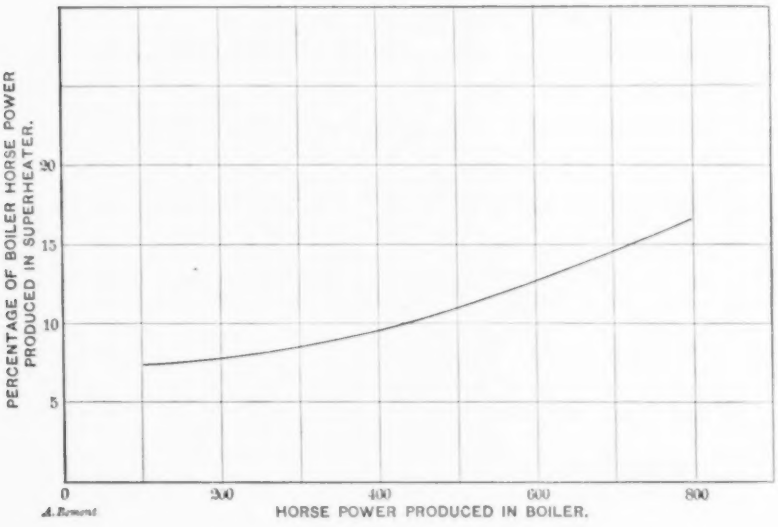

FIG. 225.

boiler to increase in much greater ratio than that in the superheater. This is well illustrated by the two tests between 100 and 200 horse-power, which gave about 85 degrees of superheat, while that at 860 horse-power only gave a superheat of about 190 degrees. There is another possible reason why the temperature elevation is not greater at high capacity; it is the possibility of water being carried from the boiler by the steam into the superheater. If this does occur, it is a case where it is desirable for a boiler to produce wet steam.

4. Fig. 223 shows relation between horse-power produced in boiler and superheater. Fig. 224 gives relation of degrees of super-

heat to horse-power produced in the superheater. Fig. 225 shows percentage of horse-power produced in the superheater of that developed in the boiler.

5. With this apparatus all of the steam flows through the superheater under normal conditions, but as the safety valves are located on the boiler drums, surplus steam escapes without being heated, which tends to cause higher degree of superheat, because the rate of flow through the heater is reduced. With sudden drop in load, superheat with the apparatus in question, has been observed to rise to 350 degrees. It has been considered that under such conditions there is danger of damage to the superheater tubes from overheating. The author is not aware that such damage has been caused, but to prevent its occurrence, it has been considered desirable by some manufacturers to also locate a safety valve on

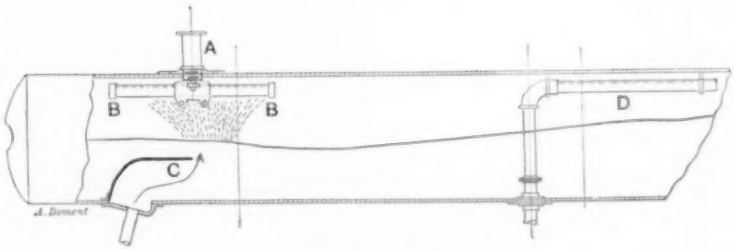

FIG. 226.—BOILER DRUM.

the superheater which will discharge at a somewhat lower pressure than those on the boiler drums.

6. One feature of this apparatus which has caused trouble, is the position of the safety valves, they being located just above the water baffle at the front of the boiler drums. Fig. 226 shows the front end of the drum. *A* is the outlet to the safety valve, *B* the so-called dry pipe leading to this outlet, *C* the water baffle. As will be observed, the steam and water from the front headers of the boiler, escape from under the baffle just below this safety valve dry pipe, there not being more than 18 inches between the normal water line and this pipe; the result is, that water is very easily carried out by the steam escaping from the safety valve. Occasionally the water line has been reduced from the middle of the gauge glass to the bottom, a drop of 6 in. occurring in a few seconds. Under such conditions the temperature of superheat rose

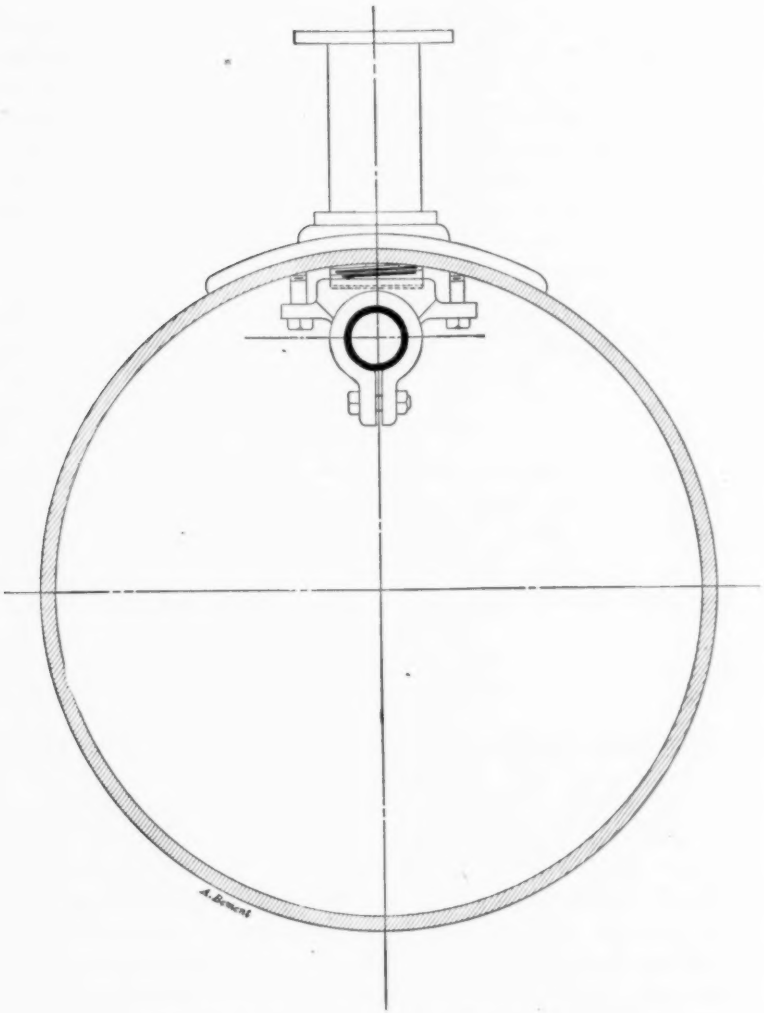

FIG. 227.—CROSS SECTION AT SAFETY VALVE.

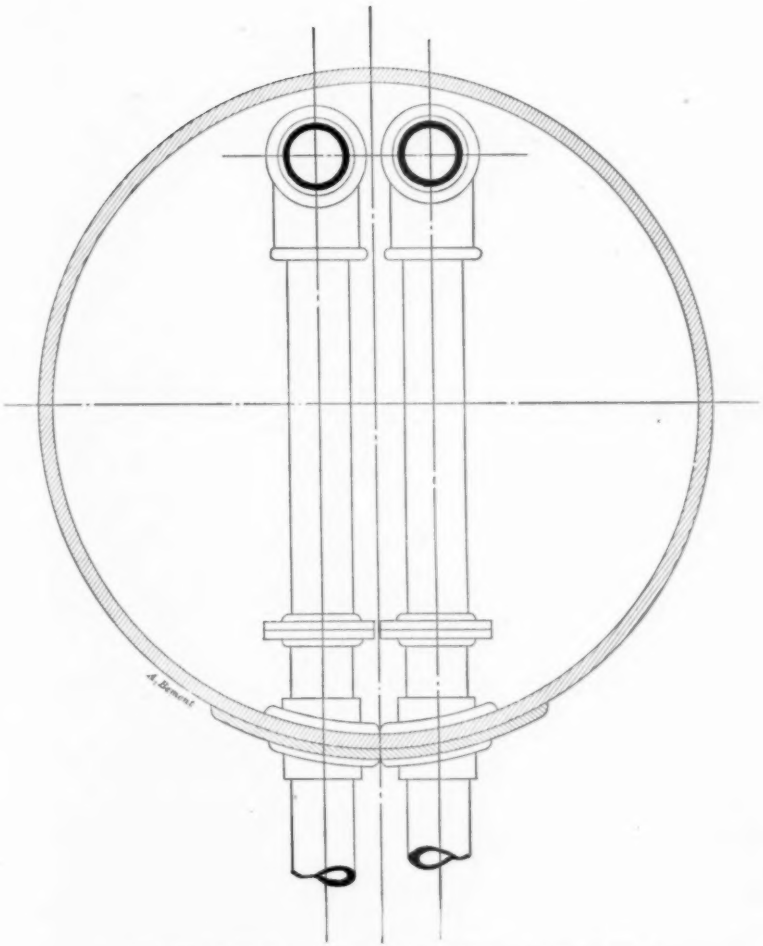

FIG. 228.—CROSS SECTION AT OUTLET TO SUPERHEATER.

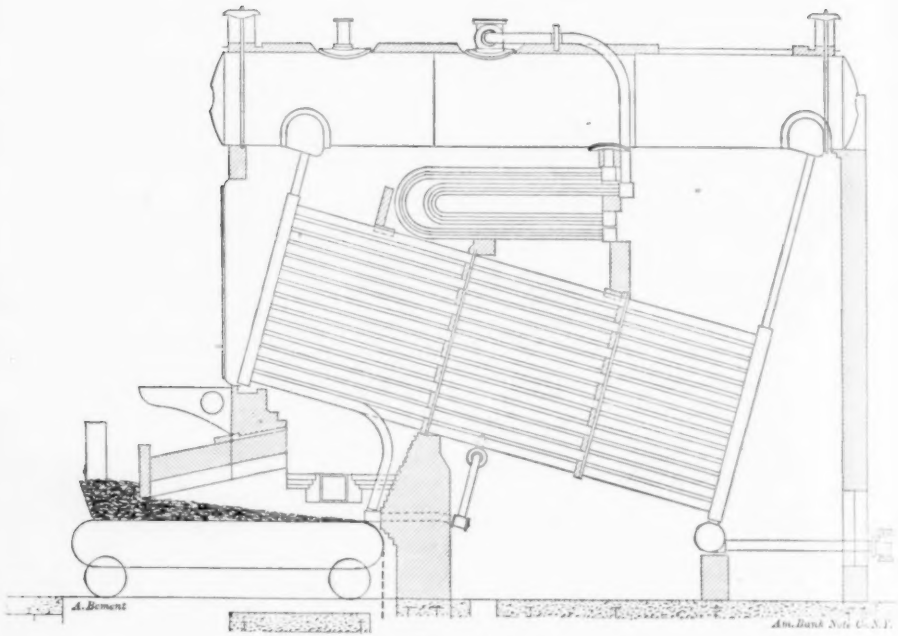

FIG. 229.

to 350 degrees. The water is indicated in Fig. 226 as rising to the safety valve pipe showing the manner of its escape.

7. Duplicate pipes *D* gather the steam and conduct it down through the bottom of the drums to the superheater which is located below. Fig. 228 is a cross section indicating these duplicate pipes, and Fig. 227 is a cross section at the safety valve. These so-called dry pipes are perforated on their upper surface with a very large number of holes of a diameter of about 0.25 of an inch.

DISCUSSION.

Mr. Geo. H. Barrus.—Mr. Bement has made certain calculations of the work done by this superheater on the assumption that the specific heat of the steam is 0.75, and he ventures the opinion that even this figure is too low. There is a deal of talk nowadays about the specific heat increasing with the pressure, and the belief seems to be quite common that the figure 0.48 ordinarily used for high pressures is incorrect. I have seen no conclusive experiments, but I am sure that if the specific heat does increase

with the pressure, the increase is much less than this paper would lead us to suppose.

I have no experiments of my own which give positive data on the subject, but I have some which may be said to give negative data, and these prove to my mind that the specific heat is nothing like 0.75. The experiments which I made are of a private nature, and I can present only the conclusions which bear on the subject. The work was done on an independently fired superheater, and consisted in measuring the weight of coal burned, the amount of steam superheated, and all the data required for determining the efficiency. If the specific heat of the steam were as high as 0.75, the efficiency of this superheater would have been no less than ninety-two per cent. of the calorific value of the fuel, and I am sure that no one would have the temerity to claim any such result. There is no reason why an independent superheater should have an efficiency greater than can be obtained from a hand fired boiler, and this will seldom show much over seventy-five per cent. The chances are that a superheater such as I tested would be much less efficient than the boiler, for the reason that it has a large mass of brick work exposed to radiation, and the surface of this brick work is far greater, and its temperature far higher, than that of a steam boiler setting of equal grate area.

If the author has any reliable data in support of his position in this matter, let us have them.

Mr. E. H. Foster.—In paragraph 5 the author speaks of a sudden rise of superheat with a drop in load.

This effect I attribute to the sensitiveness of the superheater, owing to the use of "bare" tubes. In our form of superheater we provide a heavy coating of cast iron, with deeply corrugated surface, which mass of metal acts as a dampener to sudden changes in conditions and tends to maintain a uniform superheat.

*Mr. Bement.**—It is very fortunate, I think, that Mr. Barrus makes the criticism he does; not that I agree with him, but because it illustrates how divergent are the opinions held regarding the specific heat of superheated steam, in view of the fact that his opinion differs so widely, for example, from that held by Professor Reeve and Mr. Orrok. The reason I adopted the figure 0.75 for specific heat was because these boilers supplied turbines, and the General Electric Company had used this figure. For this

* Author's Closure, under the Rules.

reason the client expressed a wish that it be employed. Mr. Barrus mentions an experiment wherein the result appeared to show that the specific heat is much lower than 0.75. The details of this would be of the very greatest importance; unfortunately, they are not furnished.

This matter is an embarrassing one. There are a number of colleges in the country which could have made valuable experiments, but it seems rather to be the disposition to accumulate apparatus than to use it. For many years chemists and physicists have claimed that specific heats increased with both pressure and temperature, but engineers have been content to overlook and ignore such views, until the matter has reached its present state.

The cast iron outer coating of the tubes used by Mr. Foster, of course, does store heat, with the result that under such conditions as mentioned, the excessive degree of superheat would not be so quickly attained, but if a calculation is made showing the heat storing capacity of this outer cast iron surface, it will be apparent that the safety valves will not have to blow very long before steam temperature would rise as high with this, as with the superheater described in the paper.

No. 1086.*

NOTES ON EFFICIENCY OF STEAM GENERATING APPARATUS.

BY A. BEMENT, CHICAGO, ILL.
(Member of the Society.)

1. This contribution was suggested by the discussion of a paper entitled "More Exact Methods for Determining the Efficiency of Steam Generating Apparatus," which was presented by the author at the previous annual meeting.

2. Efficiency in steam generation is a very complicated problem. It is a function of some constants and many variables, therefore it is necessary to consider and take into account the individual features and their influences. Especially is this necessary for proper selection or for the development of superior apparatus, and proper consideration of these matters requires a more exact definition of some terms used, than prevail at present. For example, the word boiler is employed indiscriminately to designate either the boiler proper or the combination of it with other apparatus. Likewise the efficiency of a boiler, and the efficiency which may be produced through it, are commonly considered as being the same.

The Boiler.

3. There is considerable lack of agreement as to what is implied by the term "boiler." The author does not recognize the brick walls which enclose it, the cast iron fire bars, the stoker machine or the furnace as being in any way a part of the boiler. Manufacturers take orders for, and contract to deliver under penalty, what they designate as a "boiler," and they deliver and receive payment for a metal structure, tight enough not to leak,

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

and strong enough to withstand pressure, but no fire bars, stoker, furnace or brick walls are included in the delivery. These features are often supplied by others who are not in the boiler business. Thus a boiler must have an individuality.

Efficiency of a Boiler.

4. It is a fact that boilers are of various sizes and shapes, and arranged so that the gases flow in more or less contact with them, by paths of more or less length. Now, as the purpose of the boiler in this connection is to abstract heat from these gases, it must be true that the various shapes and gas passages which are fixed features of the mechanical construction, have an influence on the amount of heat flowing to it. Therefore, this metallic structure, because of features of shape and arrangements of its parts, does necessarily produce an effect, and whatever such effect is, it is fixed and permanent, because it is due to the mechanical arrangement which cannot change, and this is what the author designates as efficiency of a boiler. Other influences may act in conjunction with it, and thereby produce a different result, but this does not change this feature of efficiency of the boiler; it is permanent, and can no more change than the mechanical form and construction. For example, if a good boiler becomes scaled and dirty, this latter influence will produce a new and inferior result, but let another inferior boiler become scaled and dirty, such influence will be the same for each. Likewise a poor fire under a good boiler will produce a poor efficiency, caused, not by the boiler, but by the fire itself; and the same poor fire under an inferior boiler will result in a lower efficiency than with the better boiler. Therefore, this superiority due to the peculiarities of mechanical design and construction cannot vary with any sort of regularity and according to any ascertainable law with every variation in kind of coal and character of fire, because this characteristic is as permanent as the rivets in the boiler itself.

5. It is useful and necessary to recognize a distinction between efficiency of a boiler and boiler efficiency, the former a constant for any particular boiler, and the latter produced by other influences which may be variable in character and effect, which, acting in conjunction with the efficiency of the boiler produces various results, and effecting the efficiency produced through the boiler.

Furnace Efficiency.

6. This has been defined as the ratio of the total heat delivered from the furnace to the total heat value of the coal consumed. This definition, however, does not take into account the temperature accompanying heat development, which is the most important factor involved, because full heat development is possible at a temperature so low as to make it impossible for any of it to flow into the boiler. Heat development can be of no use to the boiler unless it is at sufficiently high temperature for such heat to flow into it. Therefore, a definition of furnace efficiency is incomplete, if it neglects the matter of temperature. It is not only useful but necessary to recognize the distinction between heat and temperature. As in electricity, a small amount of current flows if the voltage is low, but if voltage is high, a large amount will flow in a very small wire; thus the temperature is the motive power which causes transfer of heat from the gases to the water in the boiler, which transfer is in direct proportion to the elevation of temperature.

The Furnace.

7. For a proper understanding of the principles involved, the author considers the fire grate or stoker machine as being no portion of the furnace, which is an intermediary element located between and connecting the grate or stoker and boiler; a chamber in which the gases from the fire are mixed together, wherein the process of combustion which begins at the fire is finished, and where a sufficient mixture between air and combustible may be secured, provided these ingredients are supplied to it in proper proportion. This may be illustrated by assuming a fire brick chamber in which gases of more or less completed combustion enter, together with air, then if thorough mixture be obtained with sufficient air present, combustion will be complete, and if just the required amount of air be present to fully satisfy the combustible gas, an ideal result will be attained, because full heat development will occur together with a minimum quantity of gases, resulting in maximum temperature, provided, of course, that the mixing capacity of the furnace is sufficient.

8. The proper and most convenient measure of furnace efficiency would be the composition of the gases leaving it, as determined by analysis. According to the above definition, a fur-

nace would not be a portion of some steam generating apparatus; for example, when a grate is located immediately under the boiler. In other cases, such as with many stokers, the furnace feature is only partially developed. With such fuel as anthracite coal or coke, a furnace would have a minimum value, and its maximum value would be realized with bituminous coal.

The Grate or Stoker.

9. This portion of the apparatus should supply the required amounts of fuel and air, and in such relative proportions as to cause a proper balancing of the quantities of air and combustible, or should be so designed and proportioned as to insure such result by proper manipulation, and sufficient mixture between fuel and air should occur at the grate so as not to overtax the mixing capacity of the furnace. In addition, this portion of the apparatus should remove the ash in a manner and at a rate so as not to interfere with fuel and air supply, and at the same time not waste fuel in the ashes. Thus it appears that the function of the grate or stoker is complicated. The plain hand-fired grate fulfills requirements to a minimum, while on the other hand a properly proportioned chain grate stoker fulfills requirements to a maximum extent.

10. It is not only desirable that the grate or stoker utilize the entire amount of fuel, but that it furnish a product suited to the requirements of the furnace, so that it in turn may be able to serve the boiler with a product best adapted to its especial needs.

11. Therefore it is important and necessary that the individuality of the functions and processes be recognized, because a high efficiency cannot be had from a good boiler if served with a poor fire, or from a good fire if the heat absorption is attempted with a poor boiler.

Manipulation.

12. High heat efficiency in steam generation requires, first, a good apparatus, and second, that it be properly manipulated. The faults of the fireman and furnace operator are well understood and recognized, and he is much to be blamed for poor results. But it is quite the popular thing, however, for the entire blame to be placed on the man who manipulates the fire, when in fact the apparatus is often very faulty, and in some cases very poor

results are very nearly all due to the apparatus, yet the man is blamed for the bad engineering of the maker and designer. And often an operating engineer or proprietor seeks in the fireman a degree of intelligence and knowledge of these complicated processes, much greater than he himself possesses.

13. For better guidance of the furnace operator, the automatic indication of oxygen has been considered desirable. But there is no reason for such device from the standpoint of present knowledge. Should it, however, be available and in use, and the indications were as in the two examples from the same furnace—

	No. 1.	No. 2.
Per cent. oxygen	5.6	7.6

what would the fireman consider necessary; to increase it in No. 1 or reduce it in No. 2? If an Orsat apparatus was used to learn what these indications of oxygen meant, the result would be

	No. 1.	No. 2.
Carbon dioxide	13.6	11.1
Oxygen	5.6	7.6
Carbon monoxide	0.0	1.1

Being guided by the indications of oxygen only, it might appear to the fireman that it should be reduced in No. 2, but combustion is not complete as it is; the indications are that it could be reduced in No. 1, which would not be apparent from a knowledge of the percentage of oxygen only. These two analyses were from a Hawley furnace and taken about two hours apart, No. 1 with a clean fire on the bottom grate, No. 2 with an accumulation of ash and clinker on this grate, and the large amount of oxygen was caused by the amount of coal being burned becoming less, owing to the obstruction which caused the air entering other than through the fuel being proportionately greater. Now, a knowledge of the CO_2 would have shown that there was something the matter with condition No. 2, fully as well and probably better than that of oxygen.

14. There are now automatic apparatus for the boiler room which continually indicate condition of combustion according to percentage of CO_2. The Econometer and Dasymeter are purely mechanical devices; the Gas Composimeter and the Ados apparatus are chemical instruments; the latter two make permanent records; the Dasymeter is made with and without recorder, and

the author has devised and in one case applied a recording attachment to the Econometer, which makes a record on a 24-hour circular chart similar to, and as convenient as those from the Bristol recording instruments.

15. No more instruments to indicate condition of combustion are required at present, but what is needed very badly, are some more engineers and proprietors who will appreciate and learn to use the instruments which are already available and to impress their value on the firemen by their own example.

16. When it is considered that within about five years nearly every engineering college in this country has added to its equipment either an Econometer or similar instrument, and that during this time their use has been largely discontinued, not through any fault of the instrument, but owing to neglect of the important economic feature which they represent, is an indication that the fireman is not responsible for all of the trouble.

An Efficient Boiler.

17. In the paper by the author which is referred to above, results of tests were given of two boilers differing much in efficiency. As no drawings accompanied that paper, it was not clear to what type the tests referred. For this reason, illustrations showing them are offered. These boilers are designated as *A* and *B*. Fig. 230 is an elevation in section showing both in one view. What is designated as *B* was made by changing the original gas travel by the insertion of two baffles, the location of which is indicated by the broken lines parallel with the tubes. The paths of the gases are shown by arrows among the tube surface, the heavy full one is for *A* and the broken arrows for boiler *B*; therefore to illustrate boiler *A*, it is only necessary to consider the two inserted baffles and the broken arrows as missing. This improved boiler was not originally made as shown but was produced by changing one of the *A* type. In this, the first experimental unit, the inserted baffles were of sheet steel, but with others which were changed, because of the value of this improvement, the lower inserted baffle was made with fire brick tiles, and the upper of corrugated steel plates. To insert a baffle required the removal of one row of tubes which were afterwards replaced. It was found by temperature measurements made in the interior of the boiler among the tubes, that the openings at the ends of the baffles were greater

than necessary, therefore in the change of those that followed, these openings were proportionately decreased, so that at the final outlet at the top of the tubes, this baffle was extended within two feet of the front water leg.

18. Briefly the results of two kinds of experiments are given in tables Nos. 1 and 2; these tests are given in greater detail in the original paper.

TABLE I.

HORSE-POWER AND FINAL TEMPERATURE.

	Boilers.	
	A.	*B.*
Complete Combustion, CO_2	10.7	10.3
Horse-power developed	564.3	580.4
Final gas temperature	657	469
Steam temperature, F.	379	377

TABLE II.

HORSE-POWER AND EVAPORATION.

	Boilers.	
	A.	*B.*
Complete combustion, CO_2	7.0	6.7
Pounds of the same fuel burned per hour	1,766	1,446
Horse-power	383	395
Pounds of steam from and at 212 degrees per lb. of fuel burned	7.47	9.42
Steam temperature, F.	377	377

19. In Table No. 1 the difference in efficiency appearing in the final gas temperature, and in Table No. 2 the measure of efficiency is in steam made; while boiler *B* was supplied with a smaller quantity of heat, it absorbed more than the other.

Path of Gas in Boiler A.

20. Obtaining an exact value for the temperature of the escaping gases is a very difficult matter, well illustrated by Figs. 231, 232 and 233. Referring to Fig. 231, the active path of the gases is indicated according to temperature measurements taken through hollow stay bolts in the front and back water legs of boiler *A*. The illustration shows the tubes as being absent. Fig. 232 gives measurements made across the path of the gases at the exit; the points indicate temperatures at locations shown by corresponding letters in Fig. 231. The curve of Fig. 233 gives temperatures on the vertical line *Y* of Fig. 231. The location of measurement of the final temperature for each of these boilers is indicated by the

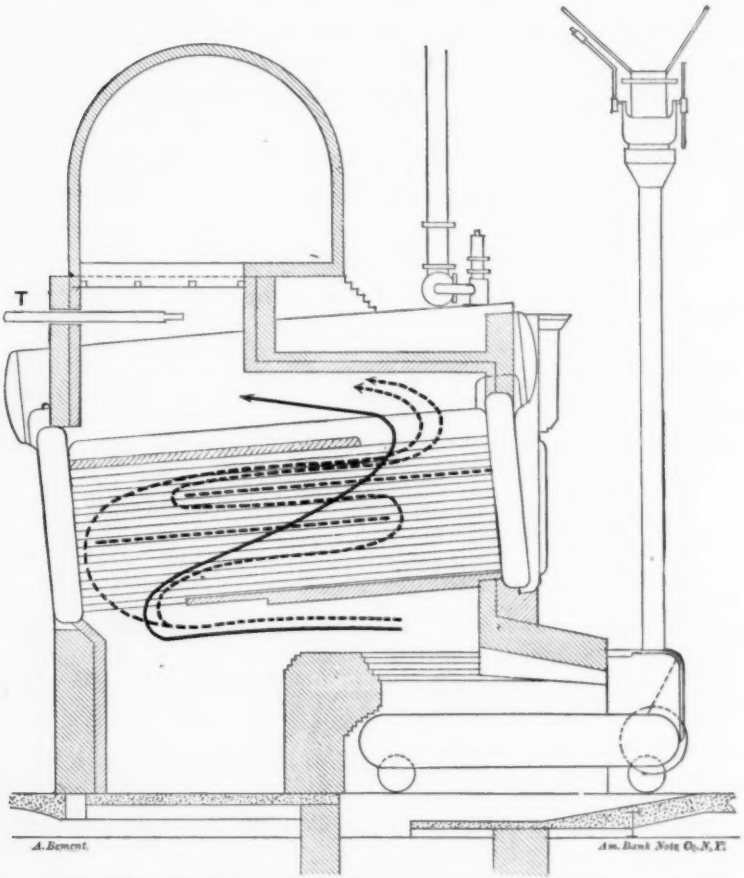

FIG. 230.—ELEVATION IN SECTION, SHOWING BOILERS **A** AND **B**.

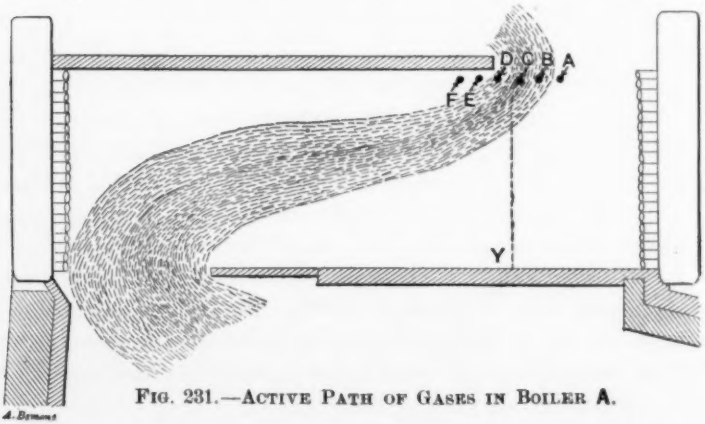

FIG. 231.—ACTIVE PATH OF GASES IN BOILER **A**.

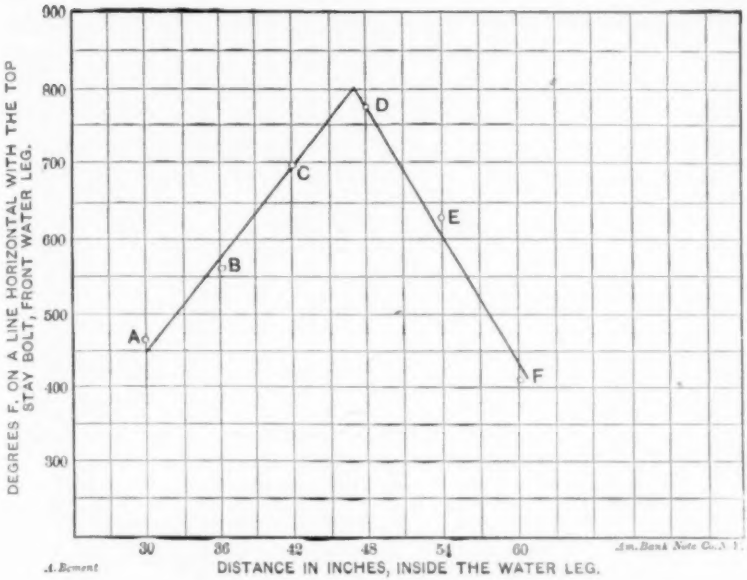

FIG. 232.—TEMPERATURE ACROSS GASES AT EXIT FROM TUBES.

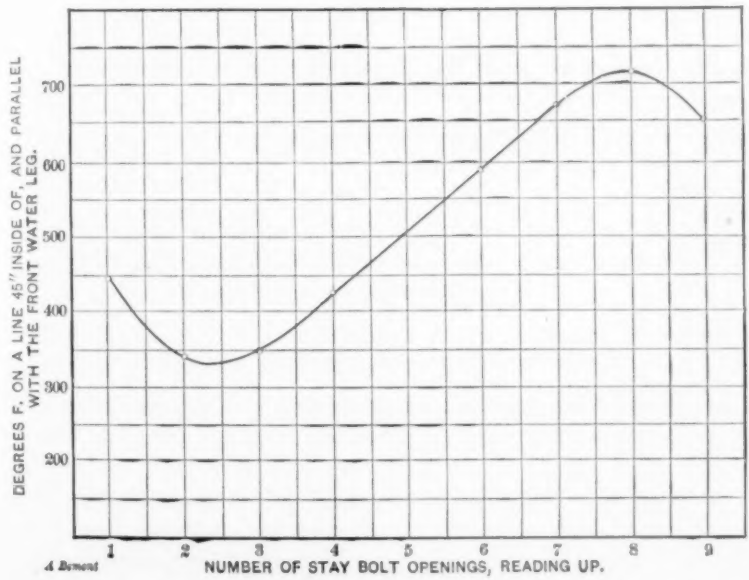

FIG. 233.—TEMPERATURE OF LINE Y OF FIG. 231.

measuring instrument shown at T in Fig. 230. This point may or may not have given an exact value, but it is similar for each, and was selected as the best location after much experience with these boilers. The path of the gases shown in Fig. 231 has been referred to as short circuiting, yet they always flow through this boiler in a direct line from the back to the front upper corner and do not pass into the other two corners at any rate of driving when suction draft is used, whatever the rate of driving may be. This may be called short circuiting, yet it is the natural path of the gases.

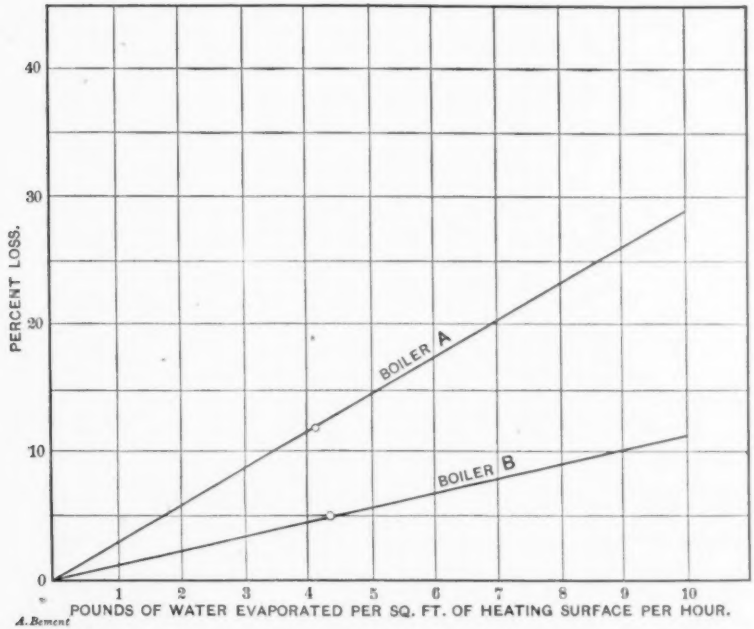

FIG. 234.—LOSS DUE TO INCREASING CAPACITY, MEASURED ABOVE STEAM TEMPERATURE.

Effect of Boiler on Final Efficiency.

21. It has been intimated that there can be no advantage in a boiler of superior efficiency over an inferior one, because of the varying effect of quality of fire, rate of driving, etc. And it will be useful to illustrate some of the more important of these effects as applied to boilers A and B, because it appears to have been intimated that these variable influences may efface the superiority of a boiler.

22. The efficiency produced through a boiler decreases with increase of capacity, on account of increasing resistance to flow of heat through the metal. Therefore, the losses in the hot gases above steam temperature for table No. 1, are indicated by the two points in Fig. 234 at their respective rate of evaporation. Now

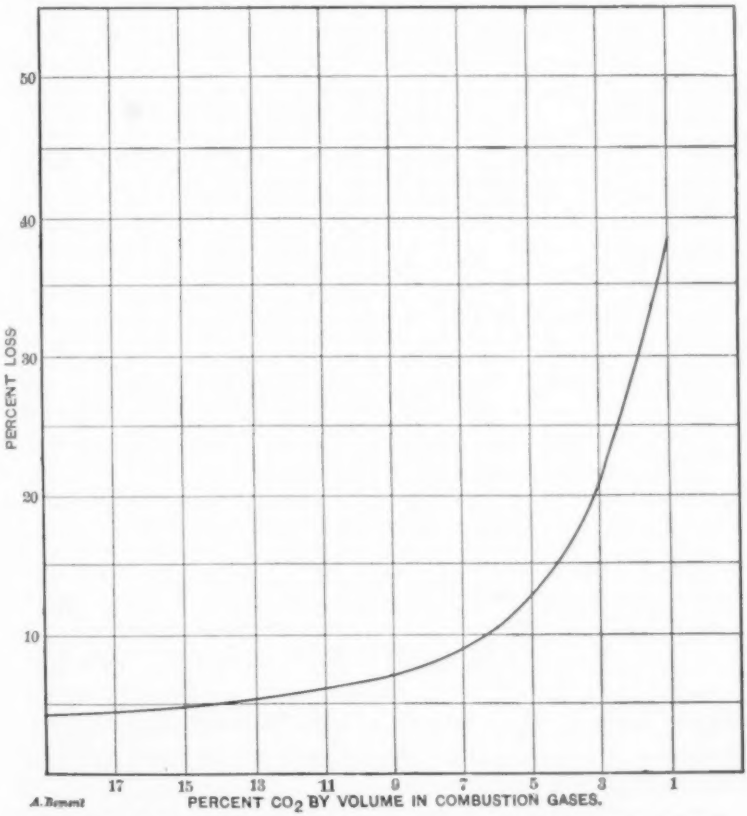

FIG. 235.—LOSS FROM BOILER **A** OVER **B** DUE TO DECREASING INITIAL TEMPERATURE, MEASURED ABOVE TEMPERATURE OF AIR.

if a sufficiently small fire be maintained under these boilers, there would be no loss above steam temperature, or, in other words, this would be a condition of zero loss above steam temperature. It does not matter that radiation from the boiler would carry away a large amount of heat, because this fact has nothing to do with the problem here presented, therefore the effect on these two boilers, caused by increasing capacity may be illustrated by

the two curves; these curves are not properly straight lines, however, but should drop off somewhat; the illustration, however, is sufficient. So it follows that the effect of increasing capacity is to produce a diverging result in favor of boiler *B*.

23. As condition of combustion has an influence on the amount of heat which will flow to any boiler, it affects the efficiency produced through it. Taking the final temperatures above the air supply of 75 degrees from Table No. 1, the curve in Fig. 235 illustrates the increasing loss caused by boiler *A* over that of *B*, with

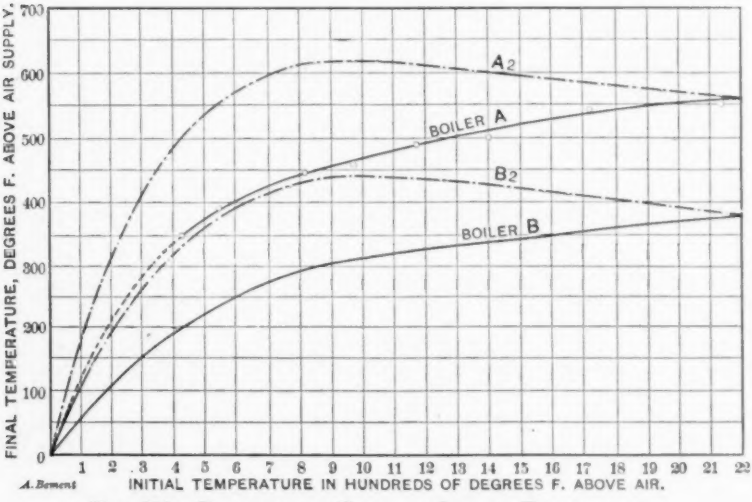

FIG. 236.—RELATION OF FINAL TO INITIAL TEMPERATURE.

varying excess of air, assuming that the final temperatures of Table No. 1 are constant for all conditions of combustion, and those capacities given.

24. The reason why constant final temperature is assumed is illustrated in Fig. 236. The curve designated boiler *A* is plotted from simultaneous measurements in the furnace over the fire and at the exit from the boiler. After the first reading was taken, the motion of the chain grate stoker was arrested; this allowed the fire to gradually burn away, which proceeded from the back of the grate toward the front, so when the last set of measurements at 425 initial and 350 final were taken, there was but a small quantity of fire present at the front of the furnace. This curve was produced by decreasing quantity of heat generated and

by decreasing initial temperature, the former due to the fire becoming smaller, and the latter to enlarged excess of air, with the result that while the initial varied by 518 per cent., the final temperature varied by only 36 per cent. It is, of course, unnecessary to state that these measurements were not made during the progress of any other experiments. The curve designated boiler *B* is so located as to represent corresponding relation between initial and final temperature for its boiler and relative condition, and is not plotted from temperature measurements, being located a distance lower on the diagram, corresponding to the difference in final temperature given in Table No. 1. These curves represent one extreme of the effect produced on final temperature by a varying quality of the fire as influenced by the two boilers in question.

25. Excess of air reduces initial temperature; for this reason, flow of heat to the boiler is decreased and more goes to the chimney, resulting in a higher final temperature. Therefore, low initial causes high final temperature, and when decreasing initial temperature is accompanied by generation of uniform quantity of heat or by increasing quantity of heat, the final rises inversely to that of the initial temperature to a marked extent. This, the other extreme, is indicated for the two boilers by curves A_2 and B_2. Taking the effect produced by varying qualities of fire and initial temperature as illustrated by the curves showing the extremes, it appears that the curve of average final temperature would be a horizontal line beyond the point of 950 degrees initial.

26. But it makes no difference whether the final temperature is based on the average as illustrated, or on either of the extremes indicated by the curves, all show the increasing value of boiler *B* over *A* as the fire becomes poorer; while, on the other hand, as the quality of the fire, or, in other words, initial temperature increases, the value of the fixed and unvarying efficiency of a boiler becomes less, not owing to any change in the boiler itself, but in the effect produced upon it by the fire. So that as the initial temperature rises the resulting efficiency produced through these boilers comes nearer together.

27. The author considers that these boilers, *A* and *B*, represent the extreme ranges of efficiency, possessed by what would be called first-class boilers, and it is his present opinion, if such or corresponding boilers be served with a very good fire, the resulting efficiency produced through them will equal a fuel saving of

10 per cent. for the better over the poorer, at a capacity of one horse-power developed per each 10 square feet of heating surface, and this would be the minimum measure of value between them. But, as the condition of combustion becomes poorer, the effect of the value of a more efficient boiler becomes greater, and a purchaser would always realize benefit from it, which would not be the case with a good furnace and stoker over poor ones, unless properly operated.

DISCUSSION.

Prof. William Kent.—"It is useful and necessary to recognize the distinction between the efficiency of a boiler and boiler efficiency, the former a constant for any particular boiler."

I have never found that such is the fact, and I do not find that anywhere in this paper the author has given any method by which we can determine what he calls the efficiency of the boiler; that is, the constant by which we can rate a boiler and say this boiler has a certain efficiency, which is a constant for this particular boiler. I do not understand how he is going to arrive at this so-called efficiency, or what the means are for getting at it. Of course, we all know what has been invariably called the efficiency of a boiler, and that it is clearly laid down and defined by the code of the Boiler Test Committee, and I do not think we really need any other definition.

The efficiency of a boiler is a figure obtained by a test, it is not a function of the boiler only, but also of all the variable conditions that exist during the test. The commonly used definition agrees with the general definition of efficiency of any machine, that is the ratio of "output" to "input." It is the same thing which the author calls "boiler efficiency" or, elsewhere in his paper, "efficiency produced through the boiler. The attempt to define it otherwise makes a definition which is out of harmony with all existing literature on the subject. A boiler can no more have a definite constant efficiency, a function of the metallic structure only, than a steam engine can; the efficiency of a steam engine varying with the load, with the steam pressure, back pressure, etc.

The results of Mr. Bement's tests are interesting in showing that the efficiency of a boiler (as ordinarily defined) is greatly decreased by short circuiting of the gases and also by excessive air supply, but this is a well known fact.

I must take exception to the statement in paragraph 22, that

efficiency "decreases with increase of capacity on account of increasing resistance to flow of heat through the metal." There is an increased flow of heat through the metal, but there is not enough heating surface to absorb the excess of heat supplied.

Mr. Albert A. Cary.—As the first part of Mr. Bement's paper is somewhat similar to my discussion of his former paper, entitled,—"More Exact Methods for Determining the Efficiency of Steam Generating Apparatus," I find that discussion almost applicable to this paper, but there are certain points which might be added to my former remarks which may prove interesting here.

Professional men have become too prone to regard, in an engineering sense, a complete steam generating equipment as a *boiler,* whereas, had they turned to their old Webster's Dictionary, taken his definition and in their investigations have given the boiler the individuality it must have in future considerations, we might find less radically bad practice than actually exists to-day in boiler design.

Webster defines a boiler as "A strong metallic vessel, usually of wrought iron plates riveted together, or a composite structure variously formed *in which steam is generated* for driving engines, or for heating, cooking or other purposes." Nothing is said in this definition of mason work, furnaces or grates.

The efficiency of this individual piece of apparatus varies with its design and also with the temperature and quantity of heat supplied to it by the furnace.

The ideal boiler shoud receive gases at the highest attainable furnace temperature and discharge them at the same temperature as the surrounding air. Barring unusual cases where an extremely bad setting might allow an exceedingly large infiltration of outside air, the gases which heat the boiler and its contained water, of course, cannot be discharged at a temperature lower than the temperature of the water and steam in the boiler. Any *great* excess of discharge temperature above this degree is due to either lack of sufficient heat absorbing surface, or else due to an improper disposition of the furnace gases as they travel from the furnace to the flue.

Carefully conducted boiler tests have taught us that in average practice, for maximum economy, not less than one square foot of water wetted boiler surface (placed in the path of the hot furnace gases) should be provided for each 3 pounds of water evaporated per hour from and at 212 degrees. This means that 2897.1

British thermal units are supposed to be transmitted to the water per each square foot of heating surface when this rate of evaporation occurs.

With the definition of individual boiler efficiency given by me in discussing Mr. Bement's previous paper, we can see that if our boiler (considered apart from the furnace) were of 100 per cent. efficiency for every 2897.1 British thermal units delivered per hour by the furnace to the boiler, we should provide one square foot of heating surface.

This assumption, of course, does not consider chilling effects from radiation nor cold air infiltration through boiler settings.

As the individual efficiency of the boiler decreases, we find that a greater quantity of heat must be supplied from the furnace to the boiler to care properly for this square foot of heat absorbing surface as shown in the following table:

With a boiler efficiency of	B. t. u.'s required.
90 per cent.	3219
80 "	3621
70 "	4139
60 "	4829
50 "	5794

Probably the principal reason for low individual boiler efficiency is the second cause mentioned above: viz., an improper disposition of the furnace gases traveling from the furnace to the flue, and this is well illustrated in the cut shown by Mr. Bement on page 626 of his paper.

In my discussion of Mr. Bement's paper at the last meeting of the Society, I mentioned my success in increasing the individual efficiency of a very similar boiler after ascertaining by careful pyrometric and other methods just where the trouble existed and what was its extent.

After the hot gases are delivered by the furnace to the boiler, to obtain high boiler efficiency, they must be guided so as to envelop and bathe the *whole* heating surface uniformly from the time they reach the heat absorbing surface until they leave it.

There must be no cold pockets nor dead spaces where the gas does not circulate freely and bodies of gas of more or less volume must be broken up and evenly distributed over the entire path in which they are designed to travel.

My attention was recently called to a peculiar occurrence in a large vertical water tube boiler where the tubes were arranged

around a central shell in such a manner as to form several clear, chimney-like openings through the coils of tubes.

These openings extended from the furnace directly upward into the hood, forming the stack opening.

Gases were discharged through these clear passageways at a temperature of about 800 degrees, while the temperature on the same parallel plane but over the nest of closely arranged tubes was less than 500 degrees. By closing these chimney-like openings, the efficiency of the boiler was appreciably increased.

I have always found, in the most efficient boilers, a gradual and more or less uniform drop in temperature from the point of first contact to their position of exit from the heating surface and a curve traced to show the gradual drop in temperature in such boilers should be uniform and continuous. Of course, such temperatures must be taken along a single line of gas travel.

Any sudden breaks in such temperature curves always indicate a point of faulty construction or bad arrangement of the path of travel for these gases.

For high boiler efficiency a high furnace temperature is most essential.

If the specific heat of the gases remained constant at all temperatures, their drop in temperature between the furnace and flue outlet might be a direct function of the boiler's efficiency.

Thus with no outside disturbing effects, we would find that with a furnace temperature of 1000 degrees above the air supply and with a flue temperature of 500 degrees above the air supply, a boiler efficiency of 50 per cent. would be obtained and if the initial and final temperatures were 2000 degrees and 500 degrees, a boiler efficiency of 75 per cent. would exist, while 3000 degrees and 500 degrees would give us a boiler efficiency of 83½ per cent.

High furnace temperatures do not necessarily produce high flue temperatures in boilers, as Mr. Bement has shown in his chart on page 630. This is due to the much higher rate of heat transmission as the difference in temperature between the two sides of the transmitting surface increases.

The necessity for high furnace temperatures is thus shown to be necessary to obtain the highest boiler efficiency.

When high furnace temperatures are obtained, the gases should be delivered to the boiler in as evenly distributed a manner as possible so as to produce no initial concentration of this heat upon a small area of the presented heat absorbing surface of the boiler.

With such high furnace temperature, in well designed boilers, there is no danger of damage occurring to the boiler, providing the boiler is kept free of scale and deposits, and there is no excuse in these days for allowing harmful boiler deposits to occur.

This accumulation of deposits inside of boilers as well as accumulations of soot, ash, etc., on their exterior may be considered as secondary causes which may affect the efficiency of boilers, but if they are properly designed and set to allow frequent and easy cleaning, such troubles practically vanish.

Regarding boiler efficiencies being affected by the quality of fire, rate of driving, etc., as mentioned in this paper, it is not rational to suppose that a boiler having its heat absorbing surfaces imperfectly bathed in the flowing furnace gases should become more efficient with a poor fire, and low furnace efficiency, than when it is operated with a well manipulated fire and a high furnace efficiency.

Supposing that a boiler is provided with a proper amount of heat absorbing surface, but the hot furnace gases it receives are sent over the heat absorbing surface in such a manner that when we make temperature measurements across any section of the path of gas travel (at right angles to their direction of flow), we find only a fractional part of such surface is bathed with these gases at their highest temperature while the balance is bathed with gases at a lower temperature. This range of temperature is shown, in one case, to vary from 400 degrees to 800 degrees in the chart (Fig. 232) of Mr. Bement's paper.

Suppose again another boiler of similar proportions, with its furnace gases conducted over its heat absorbing surface in such a manner that a practically even temperature is maintained across the entire area of a section of gas travel (at right angles to its direction of flow).

In the first case we have a boiler with only a portion of its heating surface made capable of producing a high rate of evaporation, while a lower rate occurs at all other positions, and different parts of the heating surface must have different heat transmitting values.

In the second case we have practically uniform evaporation over every square foot of heat absorbing surface which promotes the maximum possible absorption of heat under all degrees of practical furnace temperatures, and it follows that lower flue temperatures must result, as compared with the boiler described in the first supposition.

It is scarcely worth considering, the matter of forcing boilers, after the above discussion, as it is easy to see the great advantage, and higher efficiency of a boiler with its entire heat absorbing surface evenly bathed than when the hottest portion of the gases at any section of the path of flow are concentrated upon a fractional portion of the heat absorbing surface, while the balance is bathed in stagnant and slowly moving gases of a much lower temperature.

Turning now to surface considerations, Mr. Bement, on page 621, takes exception to the definition of furnace efficiency given by me in discussing his previous paper, on account of its not considering resulting temperatures.

In this criticism he seems to confuse furnace efficiency with combined boiler and furnace efficiency and he seems to lose sight of the fact that the furnace is purely and simply a piece of chemical apparatus built for the purpose of promoting and effecting certain chemical combinations between the available oxygen and the combustible constituents of the fuel, and if complete and maximum oxidation occurs in the furnace, before the resulting products of combustion leave it or its attached combustion chamber, 100 per cent. furnace efficiency is attained and as a very high furnace temperature is the invariable indication of the best furnace conditions, with such results as just described, we can safely expect a very high temperature resulting.

The theoretical elevation of temperature of the fire may be calculated by the formula—

$$T = \frac{H}{WS}$$

where T is the elevation of temperature; H, the British thermal units generated by the combustion; W, the weight of gaseous products, and S the specific heat of these products.

In the theoretical furnace of 100 per cent. efficiency, just described, all the British thermal units contained in the coal have been generated in the furnace, the same as in a Mahler Bomb Calorimeter. Again, as such perfect combustion could not be realized in the presence of too great an excess of air, chilling and diluting the combustible gases, the weight of the gaseous products of combustion must be practically at their minimum, which facts show when referred to this formula, that high tem-

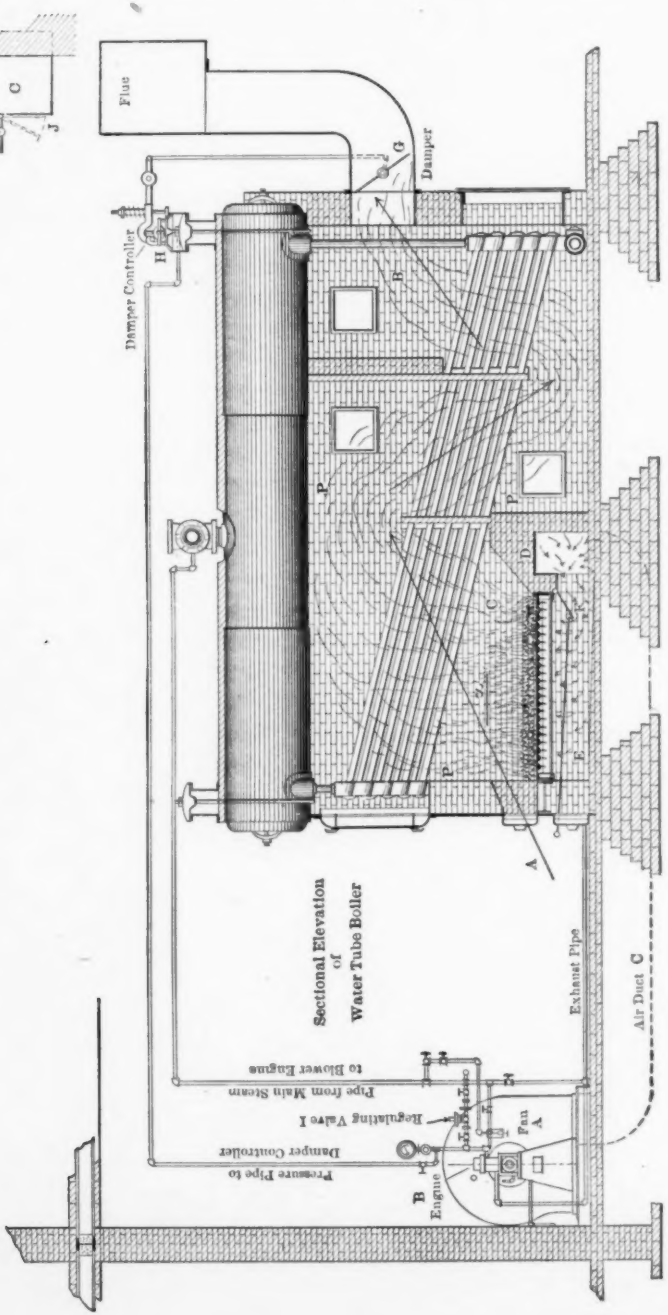

FIG. 237.—BALANCED DRAFT SYSTEM.

peratures must result, and therefore, the definition of furnace efficiency need not consider the matter of resulting temperatures.

As most grates, whether stationary, or mechanically operated, may be used and fitted into furnaces of different forms, each requires separate consideration in furnace design. I have often found in my practice, one form of grate totally unfitted for use in a furnace of the best design to obtain the highest results from

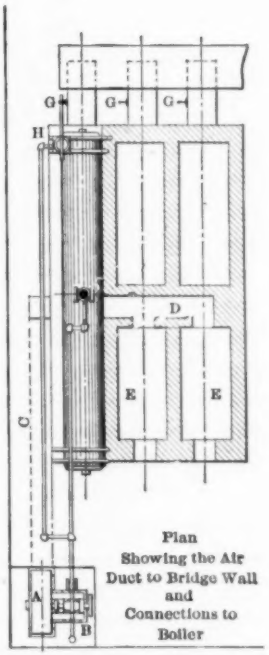

FIG. 238.—BALANCED DRAFT SYSTEM.

the particular fuel under consideration and have, therefore, been obliged to reject it for some other better adapted form.

Mr. A. J. Herschmann.—The illustrations (Figs. 237 and 238) are of the balanced draft device which we are now using in the Fuller Building in New York. Ordinarily an engineer would expect to have a very sharp blast in a chimney in order to get as favorable a condition of combustion as possible, but if you actually get about 1½″ draft in a stack this is what occurs: The air comes in at the point (*A*), and travels at the angles indicated,

passing into the stack at (*B*). Naturally there are pockets formed (*P*) in which there will be no gas at all and the gas will simply take a short cut down. These pockets will always indicate the parts of the boiler that are not active in generating steam.

The device which we employ in the Fuller Building is not a steam jet, as Professor Benjamin seems to have thought. It is a fan about four feet in diameter, which blows the air under the fire. We get about half an inch of blast under the furnace, and, by the time it passes through the coal, it has a perfect balance of draft at (*C*), so that when you open the fire door a handkerchief held in front of it will be perfectly limp. This condition is brought about by automatic movement of the damper. Naturally when the fireman opens the door there will be no cold air rushing in and chilling the boiler. With the use of this device it is possible to burn very small sized coal. For instance, if we use buckwheat No. 2, with the ordinary draft, there would be difficulty in charging the furnace; but using a blast under the fire we can send the air through so as to enable it to pass through the fuel without letting the fuel go through the grate.

Prof. F. R. Hutton.—The action of which Mr. Herschmann speaks is made particularly clear by the self-cleaning tubes which are in the direct current and where the dead pockets from the deposit of free dust are very much greater than in the direct current of the gas.

Mr. F. J. Bryant.—I would like to ask Mr. Herschmann at what temperature he forces the air in, and also how he heats it so as to prevent the cold air coming under the fire and dampening by moisture? I know there are several types of this form of furnace auxiliary used, but I think in each case they warm the air.

Mr. Herschmann.—The air is not heated at all. We take the ordinary air and send it in to the fire through the fan. There are devices, I know, where the air is heated; they naturally call for more power.

Mr. Blauvelt.—I should like to ask whether the economy claimed for the device spoken of is greater than with a properly dampered stack. Does not the expense of the power necessary for blowing the fire more than counterbalance the loss in efficiency which would be sustained by running the boiler with half an inch suction above the fire, which would give the same relative conditions above and below the grate as those described.

Mr. Herschmann.—The economy of this device seems to be remarkable. I made very careful tests and found, comparing the burning of soft coal and the ordinary stack by this device and burning buckwheat No. 2 with this device, a difference of 34 per cent. in favor of buckwheat No. 2. Against that I charged about $600 for an additional fireman, although that is not conceded by the people who introduced the device. They naturally claim that where boilers aggregating 300 horse-power had to generate 400 horse-power they should have a second fireman, whereas we only have one fireman running the boilers. So this $600 against the economy is in dispute. Furthermore, I charged about $400 on extra ashes. That also is not conceded, but on a coal bill of $12,000 we should, according to the test, have a gross saving of $3,500. Charging all these different items against the economy, we should still get about 24 per cent. clear economy.

Mr. Blauvelt.—The novelty of the plan, as I understand it, is the entire elimination of the heavy draft through this tall stack and the substitution of a forced draft; and my question was intended to be as to the relative economy of the method described. Suppose the heavy draft of that tall stack was dampered down so as to produce half an inch of suction; you would then have the same relative conditions above and below the grate as at present, except the losses due to infiltration of air above the fire and the chilling of the gases by that infiltration. On the other hand, you now have the losses due to the generation of the power expended in maintaining the half inch pressure below the grate. I should like to know which method would be the more economical under the conditions you have described.

Mr. Herschmann.—It would seem theoretically that the matter may be disputed. But I can refer to tests that I have made, which should be conclusive proof. There is this to be said, that with the use of such a device there is no infiltration of air at all because the boiler brickwork is not on a partial vacuum within and subject to atmospheric pressure from the outside. With the ordinary stack where you get about $1\frac{3}{8}$″ of draft you have a likelihood of false air finding its way into the fire. The dampers being actuated in connection with this device, as soon as the pressure goes up the damper shuts down. And so with the fan; this enables you to carry more uniform steam pressure, and naturally should increase the economy of engines and pumps.

Mr. Bryant.—I looked into this same subject in a little differ-

ent way only a short time ago. The method that I was looking up was the heating of the air before it is forced in under the grate, and from the investigation that I was able to make I found that the mechanical loss due to the blower was quite considerable—running up, I believe, to something like 6 or 8 or even more per cent. of the steam produced—and it occurred to me that it was something like the passing out of funds at one end after taking them in at the other end. I personally have been unable to see very much of a gain in the duty in the investigation that I have made, although I confess it has not been a very exhaustive one.

Prof. Kent.—It seems to me that most of these comparisons of improved efficiency are made on the principle of "before and after," that is, taking the condition before the device is put in and then the condition afterward, and comparing the results. Some ten years ago, Professor Carpenter, of Ithaca, and myself had to make a boiler test on the before and after principle, and we found after the device was put on that we had apparently a twenty per cent. gain. We reported that on the face of the returns it looked like a 20 per cent. gain, but that the result was ridiculous, for we knew that the thing could not have produced any such gain. We found that the only reason we got this apparent gain of 20 per cent. was because the "before" result was a very bad one, and the "after" result was only moderately good. We knew that we could with a horizontal tubular boiler, with ordinary firing, and using ordinary coal, have reproduced the good result; but by some accident the first test was a very poor one.

So in testing all devices, instead of making comparisons on the before and after principle, we should compare the results obtained with the device, with the best results previously recorded. For example, we know and have known ever since the Centennial Exposition in 1876, what good records are, and we know that the best records ever obtained in boiler practice with anthracite have been with natural devices (draft and ordinary firing), and not with special devices, and the results obtained in 1876 have never been bettered in any anthracite coal tests. I think the limit was reached thirty years ago and that none of these improved devices are really going to improve economy to any great extent.

Mr. Herschmann.—I would like to say that I wish to be understood as merely stating the difference in results between what we had in the building while burning soft coal and what

we got afterwards burning hard coal. I do not wish to be understood as being absolutely certain that the difference was brought about directly and solely by the putting in of this device. It was the difference between burning expensive soft coal and the cheapest grade of hard coal. It must be said that buckwheat No. 2 could not be burned with natural draft; it would, therefore, seem that credit is due to any device by the use of which it *can* be burned.

Prof. Kent.—That makes an entirely different proposition, of course. I did not understand the gentleman to say that it was simply a comparison between the two different kinds of coal. We can often effect a great saving in economy by substituting cheaper coal, and it is often necessary to put in a forced draft to burn the cheaper coal. I am in perfect agreement with Mr. Herschmann on that point.

*Mr. A. Bement.**—In reply to Professor Kent, I will illustrate the need for a method of determining the efficiency of a boiler. Let us assume two boilers of exactly the same size and design, one fired with an excellent semi-bituminous and the other with a low grade coal high in ash. It will be as easy to produce an efficiency of 70 per cent. through the one fired with the good coal, as to get 50 per cent. from the one with the poor coal, yet the boilers are exactly the same, and for this reason their efficiency cannot be in the ratio of 50 to 70, but the efficiency produced through them is in this ratio. It was not intended in the paper to offer a method of arriving at the efficiency of a boiler, but in a former paper,† I suggested certain standards such as steam pressure, condition of combustion and rate of evaporation per unit of boiler surface.

It is stated that a boiler can no more have a definite constant efficiency than a steam engine can. If this is true, then people make a mistake in buying compound engines with many valves and elaborate valve gear instead of those with only one cylinder and one valve.

I am very much pleased that Mr. Cary agrees with me to such an important extent. There is one point in particular, however, regarding which we appear to be not in agreement. This is the feature called furnace efficiency. It is true in the

* Author's Closure, under the Rules.

† "More Exact Methods for Determining the Efficiency of Steam Generating Apparatus," Vol. xxvi., page 418 of the *Transactions*.

sense used, that a "furnace" is a chemical apparatus, and as such its office is to oxydize fuel, and if oxydation, or in other words, combustion is complete, then from this point efficiency of 100 per cent. would result, but the object of the chemical process is the production of heat which is required to perform a certain work, consisting in this connection of its flowing into the boiler, which it cannot do unless developed at a sufficient temperature. The simple chemical process in itself has no useful value, therefore there is no object in defining a measure of efficiency for a process which does not produce a useful result. It necessarily follows that the temperature is a function of "furnace" efficiency, or to be more exact, of the heat producing process. He is quite in error in supposing that I seem to "confuse furnace efficiency with combined boiler and furnace efficiency." My contention was, that there is no advantage in completely burning the fuel if it does not produce some useful result. It is true that if combustion was complete with no air in excess, that "furnace" efficiency would be 100 per cent., even if the gases escaped directly to a chimney, but there would be no advantage, because there would be no boiler to heat, and if the only object was to completely burn the fuel, it would be more convenient to have an excess of air, therefore we cannot properly attempt the definition of "furnace" efficiency unless the object of the combustion is taken into consideration.

In reference to the "before and after" principle of tests mentioned by Professor Kent, it is well to keep in mind the fact that a saving cannot be effected in a case where a loss does not occur, and that if in comparison something is superior, that to which it is compared must be inferior, and in this connection, it would have been more to the point in the case of boilers A and B which I mentioned, if the comparison had been made from the standpoint of the inferiority of A rather than the superiority of B. In such a case as this mentioned by Professor Kent, where the object was to determine the effect produced by some device, it is very important that proper pains be taken to develop the full possibility of the condition being studied, which it would appear had not been done in the first one of the two tests. One of the important causes of confusion which prevail regarding boiler performance, etc., is that people almost invariably suppress the figures when the results are bad. Boiler manufacturers make a point to publish their good tests, and users are ashamed to let

their neighbors know how poor their practice is, therefore the impression is liable to prevail, as has been intimated, that because certain results were secured at the Centennial Exposition, they apply to general practice.

No. 1087.*

THE FUNCTION OF LABORATORY COURSES IN THE CURRICULUM OF ENGINEERING SCHOOLS.†

BY CHARLES EDWARD LUCKE, NEW YORK, N. Y.

(Associate Member of the Society.)

1. Instruction in engineering by the experimental method through the medium of laboratories is so generally accepted as necessary to the training of professional engineers that at this time it needs no defense. The details of execution of such laboratory instruction and the exact results to be attained which should control the executive element are, however, not the same in the different technical schools in this country. This fact was brought out most forcibly during the past summer by the numerous visiting engineers and investigating technical committees from abroad attending the St. Louis Exposition. Conversation with these gentlemen and later statements in their printed reports and papers show that while deploring the lack of uniformity of technical instruction in general in their own countries and admitting, in most cases, the superiority of the American system, they question the advantages to be derived from our admittedly magnificent laboratory equipments, basing the judgment, of course, on their observations. The unlike methods of handling laboratory apparatus in use in our different institutions, and the variety of professed aims for that part of the instruction dependent on this apparatus, which makes the European doubt that advantages are derived proportionate to the expenditure involved, is at least worth investigation.

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* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussion on this general subject consult *Transactions* as follows:

No. 582, vol. 15, p. 655: "Technical Education." Eckley. B. Coxe.

No. 850, vol. 21, pp. 646–1114: "Education of Machinists, Foremen and Mechanical Engineers." M. P. Higgins.

2. Just what benefit is to be derived from the use of laboratory apparatus is a question that must be answered, and in the answer will be found the method of instruction, based on that apparatus, conducive to the student's greatest good. It may be that this greatest benefit to the student can be obtained only with the greatest effort on the part of the instructing staff, which would be the case without apparatus, or it may be that through the medium of apparatus the professional instruction can be simplified and conducted by fewer and less able men than would otherwise be necessary. In reference to this relation of instruction and apparatus to the student, Professor Armstrong has said: "What the colleges (American) are suffering from very largely is the great over-provision of appliances and under-provision of teachers and of well-prepared students. I think if they scraped the greater part of the provision and obtained teachers who could develop more individuality in the students probably there would be a great improvement in the output." Naturally our first impulse on reading such an opinion is to deny the existence of the fault as Professor Armstrong sees it, but there is in the statement the keynote of the whole situation: for he maintains that the aim of the instruction is to develop individuality in the student, denying only the efficacy of our means.

This same aim he again states in a different form as follows: "We should adopt the plan of training students from the research point of view, from the point of view of developing their thinking power and their originality." Here again we must agree with Professor Armstrong, but again likewise question our methods when judged in the light of the results as observed by more or less disinterested and experienced engineers and teachers. American methods are criticized by another Englishman, Dr. Walmsley, who says in many cases the training is too superficial and too apt to overload the student with a large and confused assortment of facts instead of training him in principles. That so many able Europeans have found the same thing in varying degrees, viz.: that our American technical schools, though provided with unexcelled laboratories, machinery and instruments, still fail to make the best use of them in the field which they are best fitted to cover—the development of individuality and originality—but tend, on the contrary, to superficiality or the teaching of isolated facts makes it desirable that the question be discussed among ourselves. It is for the purpose of bringing out the opinions of the

members of this Society that this paper is presented, to the ultimate end that evils, if there be any, may be corrected, or if found not to exist that those who are engaged in the work of instruction may be reassured.

3. Difference in methods is entirely justified, if such differences correspond to similarly different aims. So that, provided each institution employs a method consistent with what it conceives to be the aim of the instruction, lack of uniformity of method throughout the country does not prove a failure to make the most of opportunities, on the one hand, or a failure to understand the duty of the institution to the student on the other. The real question to be investigated and discussed is not so much the determination of the best method, for there is probably no single best method, but rather:

(*a*) What should be the aim of the technical work dependent on laboratory instruction?

(*b*) Can there be more than one legitimate aim in our technical schools?

(*c*) What should be the method of such instruction to best accomplish the aim and justify the use of the machinery and instruments of a laboratory?

Of course the aim of all technical education is likewise the aim of each course of instruction, but besides such general aims as are thus inclusive of everything, there are other subsidiary, or rather more concrete objects to be attained by each course or method. The particular object of any one course may not include that of another course, therefore to state the aim of experimental laboratory instruction in such general terms as "to educate engineers" is practically to say nothing. The specific aim of such instruction must be more concretely stated to permit, by analysis, the determination of the method of conducting the instruction. When so stated it must necessarily include possibilities not covered by text-book courses, lecture courses or others variously defined. Before, however, the concrete expression of an aim is attempted, definite results which might be obtained by the student through such instruction over what he would acquire without it will be most valuable in justifying the employment of this elaborate and expensive equipment. In addition to the specific results which might be attained by apparatus instruction to the exclusion of other methods, there may be results common to all methods which are merely intensified by the laboratory, and

these are also valuable assets in favor of the existence of the laboratory.

4. The object of all professional engineering schools is to send away thinking men with the equipment to become good engineers. A good engineer is a man who can do things in a certain field of labor rather than a man who knows a large number of facts in that same field, but is unable to use them. The technical school *can* teach students facts, and *may* train men to do and to think, but may fail utterly to do so. Much depends on the student's physical endowment, more depends on the methods of instruction, but most on the characteristics of the instructor and his individuality. This applies to such parts of the course of instruction as are dependent on the laboratory apparatus. If, by the use of apparatus, the student can acquire more than without it well and good, but the use of that apparatus will be better justified if, in addition, the student can be made to think more clearly, more boldly, more originally than without it, and finally when through the use of that apparatus he can be taught to do things, then not only is the use of apparatus justified, but it must be regarded as indispensable, be the cost what it may. However, it is not sufficient that the above results are merely possible; it is even more necessary that the results be actually attained. If a result is possible, it can always be attained by proper methods.

5. For convenience, the possible results of experimental laboratory instruction are grouped under four headings, which may be more or less mutually inclusive.

(1) "To teach the student how to make the standard commercial tests." This means such tests of standard apparatus as he may be expected to exactly repeat in his future professional life, and most of which have been standardized by committees.

(2) "To judge and criticise existing data on record in books and journals." Practically all the useful data on engineering subjects have been determined by experiment and are the result of observations on apparatus under definite conditions. By the student's experimental work he may become familiar with the measuring devices in use, the limits of their accuracy, the extent of a man's possible personal error, the great dependence of results on conditions, and may learn caution in applying results, determined under one condition, to a case surrounded by totally different conditions.

(3) "To judge the limits of practical performance for ma-

chinery of all classes and to know to what degree losses of energy may be reduced or results desired improved by perfection of design." This is the inevitable result of comparing actual performance of machinery with what is theoretically possible under ideal conditions. Sufficient experience along these lines will prevent a man from attempting the impossible, as well as make him impatient with results below the normal.

(4) "How to approach a new problem, and by the application of a method of attack to make certain either a solution or the impossibility of solution." This is commonly called research, and while success may seem to be dependent to some extent on the mental endowment of the man, yet any one by drill under proper direction may learn systematic attack, and at least improve the chance of success. A new problem is never new in all its elements; there will be some features which have appeared before; so by careful analysis a broad new problem should be separable into a number of simple elementary problems, some of which are already solved, and which together will determine the solution of the broader original problem. Some of these problem elements will be solvable by computation, but generally there will be others which must be solved by experiment. By sufficiently detailed analysis these experimental elements may be reduced to such simple observations as are possible with the instruments available. This analysis and partial solution of the elements of a broad problem will call for the application of the whole experience and previous knowledge of the man, as well as his power to organize and execute.

6. All the methods in use in conducting laboratory instruction may be placed between two limits. At the lower limit there is provided in connection with each piece of apparatus a complete description of one determination that can be made by the apparatus, including instructions for manipulation, a list of data to be observed and the computation of results therefrom, all illustrated by an example. There is practically no thinking required, and about all the student acquires is skill in manipulation, aside from a few facts on the results.

7. The method which may be placed at the upper limit is pure research, in which the student is given a problem to solve experimentally and receives no further help. The method of attack, the data, the selection, use and manipulation of apparatus and the computations involved, are all left to be determined by the stu-

dent, without assistance but without restriction. An illustration of such a problem is one given by the world to all engineers for solution: "Is the gas turbine a possibility, economically and mechanically?" Between these two limits the methods in use leave to the student some unsolved elements, and require more or less thinking, and as a result will require more or less time on his part. Frequently there is a tendency to save the student effort by the use of a printed report blank, containing a list of all necessary observations, and some unnecessary ones, with a set of calculated results, and instructions for obtaining them. One of these taken from the publications of one of our well-established technical schools is given below.

Test of Gas Engine Directly Connected to Dynamo.

(1) Dimensions in inches.
(2) Length of test in hours.
(3) Number of revolutions made by engine shaft per minute.
(4) Number of explosions per minute (right-hand cylinder).
(5) Number of explosions per minute (left-hand cylinder).
(6) Cubic feet of gas consumed per hour.
(7) Temperature in degrees Fahr. (gas at meter).
(8) Temperature in degrees Fahr. (exhaust gases).
(9) Pressure of gas at meter in inches of water.
(10) Jacket water, weight in pounds per hour.
(11) Jacket water, temperature at inlet in degrees.
(12) Jacket water, temperature at exit in degrees.
(13) External current from dynamo, ampères.
(14) External current from dynamo, volts.
(15) Barometric pressure in inches of mercury.
(16) B. t. u. per cu. ft. gas at normal pressure, 60 degrees Fahr. by Junker Calorimeter.
(17) Indicated horse-power.
(18) Electrical horse-power, external circuit.
(19) Cu. ft. gas per i. h. p.
(20) Cu. ft. gas per e. h. p.
(21) Ratio of electrical to i. h. p.
(22) Heat balance in per cent. of the total heat of combustion.
(*a*) Heat equivalent of the indicated work.
(*b*) Heat rejected in jacket water.
(*c*) Heat rejected in exhaust gases radiated and unaccounted for.

Methods to be Employed in Calculating the Results.

Items 1 to 16 contain data observed during test.

(17) Let P = the mean effective pressure in pounds per square inch

l = length of stroke in feet.

a = area of piston in square inches.

n = explosions per minute.

Then:

$$\text{Horse-power for one cylinder} = \frac{P\,l\,a\,n}{33{,}000}$$

Obtain the horsepower for each cylinder separately and add, in order to obtain the total horse-power.

(18) Let I = current ampères.

E = e. m. f. volts.

$$\text{Then electrical horse-power} = \frac{746}{I\,E}$$

(19) Divide Item 6 by Item 17.

(20) Item 6 ÷ Item 18.

(21) Divide Item 18 by Item 17, etc., to the end.

8. There is no definite object stated for the test which, therefore, can mean to the student only one thing, viz.: A gas engine test consists in the running of an engine fitted with certain instruments, and the finding of numerical values to fill in these blank columns, and without blank columns to fill in no test is possible. Aside from the fact that this report blank in the student's hands leaves practically no thinking for him to do, there is danger that it will do him positive harm by giving him incorrect ideas. That there is more than one kind of useful test possible on a gas engine, and that a great number of useful facts and basic principles on the characteristics of this method of converting heat of combustion into work can be discovered with this engine as a piece of apparatus, may escape him entirely. There are many definite and important problems which may be solved by this machine, and if each were treated as a problem to be attacked by the student and analyzed into measurable elements there would result not only the acquirement of the facts as before, but also through success in systematic solution the experience would inculcate self-reliance in the ability to succeed again.

9. Every problem to be solved in the laboratory is as much a mystery to the student as the design of a propeller to drive an

airship at a given speed is to the average practical engineer, and this is all the more a reason for giving him the problem and an analysis method of attack which brings success to the average research problem.

10. There is another point of view from which the laboratory method of printed report blanks carrying lists of desired data in tabular form may seem desirable, and that is the relief of the instructing staff. Since there is so little for the student to do except read a pressure gauge thermometer, or do some other similarly simple thing, and to compute the results by dividing column X by column Y on the form, he will have comparatively little trouble and will require but little attention. By reason of uniformity of material, the report in the instructor's hands can be checked and marked on the numerical results alone, whereas, if some method of individual analysis leading to attack and solution were required and made the primary object, the numerical results, which in unskilled hands are useless anyway, assume but little importance. In this case the report must be carefully read by the instructor and judged on the ability shown in handling the problem, rather than on the numerical results.

11. Again, by recording so much for the student he is not called upon to use any previous information or apply principles derived from reading, from his class work or his personal experience, nor must he seek and use for comparison any similar data. He does not judge the accuracy of his results, nor discover the dependence of his results on conditions. These conditions for the important results in the test cited would be: Kind of gas, compression, initial pressure and temperature, proportion of air to gas, leakage of piston and valves, governor action, ignition point, back pressure, speed and many others. Thus one piece of apparatus may then be the means not only of adding to the student's fund of facts to an almost unlimited extent, but also by putting the work in the problem form, basic principles applied in the machines can be taught, as well as the experimental method of solving problems.

12. If the aim of the instruction be a knowledge of how to conduct commercial tests, a report blank containing information agreed upon as necessary by committees is quite sufficient and eminently proper. At the same time, however, should it be desired to teach no more than this, a very limited amount of apparatus of any design, whether in good or bad condition, would

suffice. A student can be taught just as well how to conduct a standard boiler test for capacity on a worn-out or old type of boiler as on a new one, or can be taught how to handle indicators and brakes or find engine capacities on old or small engines as well as on new and large engines of refined design. The solution of experimental problems to teach a method of individual solution and underlying principles of machine operation requires, however, the best of apparatus. Imagine a test to determine the economy of reheating steam in receivers of compound engines being conducted on leaky or small engines, or the relative economy of multiple and single expansion of steam in a laboratory, with only simple engines and no condensers. These things are ridiculously impossible, and yet they are useful fundamental problems in the economic use of steam, such as should enter into the course of instruction for mechanical engineers everywhere; and they offer good laboratory problems for experimental solution if the apparatus is available.

13. This research, or successive problem solution, method of conducting laboratory work with apparatus of the best seems to promise the student training of the kind that counts for success as an engineer whose whole life is one long succession of meeting and solving problems. The method may differ from what has been defined as the upper limit of methods—pure research—in giving to the student some real help, but not enough to deprive him of the necessity for good vigorous thinking with the pleasure and sense of power which results from success attained after effort. The best form for this help to take is the assignment of a definite piece of apparatus for the solution of every problem and the fitting of that apparatus with the necessary instruments, the manipulation of which may be explained, but no more. This should not, however, limit any apparatus to the solution of but one problem. One attempt to apply this modified research method made by the writer, in the experimental mechanical laboratory of Columbia University, has met with considerable success with all but a possible 10 per cent. of the students, who seem to fail utterly to get anything out of it.

14. For each afternoon's work per week in the laboratory there is assigned one lecture per week, at which the problems to be solved are assigned and the method of attack explained and illustrated by examples. Before performing the laboratory work the student must prepare for himself the first three of the seven paragraphs in his report, which are given below:

Work to be Done Before the Experiment.

(1) Concise definition of the problem to be solved. This is given by the instructor, and in practice often includes several problems more or less interdependent to be solved; in this case each being stated separately.

(2) Justification of the problem. By an examination of the problem in the light of experience or theoretic principles the nature or magnitude of the results demanded by the problem will be found to be wholly or partly impossible to predict, and hence must be measured. This is intended to induce the preliminary application of a man's whole fund of experience to the case in hand and to assist, by so doing, in the solution.

(3) Analysis of the problem in the light of available apparatus and basic principles, to such detail elements that it becomes possible to set down a list of observations to be made and a list of assumptions or constants needed and where they may be found.

After the completion of this examination of the problem the student is ready to begin experimental work, and he is not permitted to do so until he has handed in his preliminary report with the study of the subject as explained.

In the laboratory his attention is devoted to finding the answer to the problem rather than filling in columns or wondering what to do with the reading of some thermometer, or what good the reading is after he has it. This leads naturally to the rest of the report, which contains paragraphs each with a guiding title as follows:

Work to be Done During and After the Experiment.

(4) Results of the observation with as much of the method of making them as is necessary to judge of their quality or limits of application under new conditions. This is really nothing more than a brief summary of the actual experience in the laboratory.

(5) Answer to the problem, using the observations and assumptions found by the analysis to be necessary. Where the answer involves a variation of one thing with anything else the nature of the variation is indicated by plotted curves.

(6) Accuracy and errors. In this part of the report it is freely acknowledged that nothing found experimentally can be absolutely right, but results may approach accuracy more or less. Three errors are to be considered:

(*a*) Apparatus error, due to unfitness or lack of adjustment as affecting observations.

(*b*) Personal error in all work.

(*c*) Resultant error due to accumulation or neutralization of many elementary instrumental and personal errors.

(7) Interpretation of results. The full significance of the results are not always apparent by mere statement, but require some study to clarify. The interpretation usually takes the form of an explanation of the results, or the probable reasons why things were as they were found.

15. When by this method the student has arrived at an answer to his problem he is immediately conscious of success, and not only confident of the results for the next case, but eager to attack anything. He has thought out the basic principles involved in the problem, found for himself its elements and made observations which he himself decided were necessary and obtained generally, in spite of the stress of noise, heat and other annoyances.

16. The working out and interpretation of results calls again for application of all the principles known and comparison with results of other experiments, and might well be the daily experience of any of our best engineers, while the clear writing of a concise report on the whole process is likewise no more or less than common engineering practice. After the solution of some fifty or sixty such problems the student is ready for his thesis problem, which now has no element of mystery, but is a simple, straightforward process, different in detail from anything done, yet no more different than the regular problems were from each other, and again duplicating as nearly as possible conditions to be met in future professional life. In this work, however, the detail of apparatus is a new element and practically the only one, but the thesis is usually, however, from its magnitude, or large sweep, more difficult of solution, and calls for all the preparation, skill and power of organization available. When not degenerated into little more than a commercial test, this thesis is the culmination of such research or experimental problem course of training as has been described. When of this nature it merits the commendation so universally bestowed by visitors from abroad. Professor Cormack, referring to this thesis work, said recently: "I think it a most excellent training. It draws out the student, it compels him to think for himself, to marshal his forces to look up references, to compare the work which has been done by other people on the

same subject, and altogether it is very valuable as a training, even if the thesis itself cannot be regarded as a considerable contribution to existing knowledge." Here, it is the training that results from the application of the method to the solution, rather than the results, that calls forth praise from Professor Cormack, and the same thing may be true of the whole laboratory course if conducted along similar lines; the necessary gradation of work to meet the student's ability being obtained by intelligent selection of problems offering gradually increasing difficulty. As an item of interest a few of the many problems used in the regular work at Columbia are given below:

(1) Determination of the coefficient of contraction for a rectangular weir.

(2) Variation of contraction coefficient for weirs, with rate of flow, rounding of sill and end contraction.

(3) Variation of the corrective factor for pressure gauge with the scale.

(4) Variation of corrective factor for transmissive dynamometer with speed and load.

(5) Relation between mechanical efficiency of a steam engine and load.

(6) Variation of pump duty with initial steam pressure at constant head and constant steam pressure with variable head.

(7) Volumetric efficiency of pump cylinder and its dependence on piston speed.

(8) Pressure drop through steam ports as a function of steam velocity.

(9) Variation of actual water rate with load in throttle governed, as compared with cut-off governed, steam engines.

(10) Variation of actual water rate at constant load with initial pressure and vacuum.

(11) Determination of initial condensation for simple compound and triple engines and its dependence on cylinder temperature average.

(12) Gain in economy by re-heating steam in receivers of triple engines.

(13) Comparison between cylinder losses in compound engines with and without receiver.

(14) Determination of piston acceleration of triple expansion duplex pumps.

(15) Variation of m. e. p. and economy with load in gas engine.
Variation of m. e. p. and economy with point of ignition.
Variation of m. e. p. and economy with mixture.
Variation of m. e. p. and economy with back pressure.
Variation of m. e. p. and economy with jacket temperature.

(16) Gain in economy of two stage air compression by intercooling over non-cooling.

(17) Variation of capacity of steam traps with initial and back-pressure.

(18) Variation of turning effort of engine with load.

(19) Variation of efficiency of steam boilers with capacity and rate of combustion.

17. To sum up the situation discussed, there are different methods of executing laboratory instruction in engineering schools, and these range from the complete written-instructions method, which might be carried out by any intelligent man, to the pure research method, in which a problem is assigned and no assistance given for solution except facilities of laboratory and library. Equipment for such laboratory instruction is also quite various in kind and excellence, but on the average represents large outlays of money for installation and maintenance.

18. It is difficult to see how all the schools with variety of apparatus and method of using the same can accomplish the same ends, and it may be that much of our apparatus is useless, as charged by some English critics. From the discussion, however, it does seem that the aim of the instruction, or the object to be attained in the student, may justify both method and apparatus, and that old, worn or small pieces will suffice when the aim is to teach the commercial tests, in which case also the complete printed report form is satisfactory. When, however, it is the aim of the instruction to make useful engineers, in the highest sense, by sending out bold and clear-thinking men, well equipped with the fundamental principles and their application, then the modified research method in some form is absolutely necessary. In this case the great range of problems and variety of the scientific foundation material make the most complete laboratory, none too good nor need any part of it lie idle for want of usefulness.

DISCUSSION.

Mr. A. Bement.—This is a paper of the greatest importance, and Doctor Lucke has certainly performed a valuable service in bringing this matter before the society. Educators would probably deny the broad assertion that the tendency in an important measure aims toward an effort to supply the student with a stock of ready made information which, with slight modification, may be employed to settle problems as they are presented, rather than to educate in the broader and proper sense, and there probably is more truth in this than we realize.

It is certainly more important to develop in the student the ability for clear and correct thinking and method for analysis, than to supply him with a store of information and a limited amount of experience which can have no exact application in subsequent practice.

It is no doubt true that many colleges are expending funds for elaborate laboratory equipment which could be more profitably employed for better and more extended instruction. There is probably a greater competition among American colleges toward the acquisition of laboratory equipment than for excellence in teaching method, and I suppose it is easier to obtain funds for new apparatus than for proper increase of professors' salaries or for additional instructors or assistants. I have known of an expensive instrument being consigned to the lumber room, simply to make place for a different form which served the same purpose, when additional expenditure for the instructing force was very much needed.

It is my opinion that the development of research method, not only for instruction but in subsequent practice is of the very greatest importance for successful engineering work, and as Doctor Lucke desires expression of opinion, I endorse his argument.

Mr. Hugo Diemer.—In considering what should be the function of laboratory courses in engineering schools, the broad view is the one which considers not only such courses as they are given at the very best of American technical colleges, but of the work as conducted in the average engineering school. The broad view will also consider the function of the engineering laboratory as part of the training, not only of young men who are to become professional consultants, but also of the average technical gradu-

ate, who fills a salaried position in some industrial enterprise. It must always be borne in mind that the percentage of really brilliant men with creative ability is not much greater among those who have been fortunate enough to secure a technical education than among the average run of mankind.

My own experience in employing technical graduates has been that our foreign critics are justified in their belief that there is a certain lack of thoroughness in the work of the average young graduate and a tendency to superficiality. I believe the main cause for this is the fact that the average engineering school crowds its students too much, and the students' tendency is to turn in any kind of work so long as it is good enough to be accepted, because he cannot do all that is demanded of him and do it all well.

The average student realizes while in college that it is well nigh impossible to do thoroughly and conscientiously all that is asked of him, and in most colleges it is not expected that he will do thoroughly what is asked of him.

This crowding, when applied to laboratory courses, is likely to be particularly harmful. Instructors, in their anxiety to make a good display with competing schools, are apt to set a section of students to work at thermal analyses of steam and gas engines with insufficient preliminary laboratory preparation. Before such analyses are undertaken the student should be completely conversant with every mechanical detail of the apparatus he is testing and the instruments he is using in making the test, and such familiarity requires far more time in actual laboratory work than is usually assigned in engineering schools.

The artificial atmosphere of the School of Arts in the average university is apt to spread to the engineering school, and affords the temptation to the instructor to conduct his classes prematurely into ultra-scientific and abstruse fields. As a consequence he will have them undertake work which, while offering opportunities for thoroughness and development of individuality, are beyond the capacity of mind or physical endurance of the average student. There are many homely, plebeian experiments that are within the student's capacity which will develop his thoroughness and individuality just as well as these higher and more abstruse and elaborate works, which should not be undertaken before a thorough foundation has been laid for them. The fact that a student has been one of a group who have filled out blanks for a thermal analysis in no way relieves him of the responsibility of being able

to set a valve, to adjust a governor, to regulate a carburetor, or to find a knock or leak.

The permanent results gained for the student are apt to be much better with experiments that are not elaborate. This is true not only of the thermodynamic laboratory, but of the work done in the machine shop. Time is well spent in turning out nothing but wasters or useless pieces of metal if, during the process of metal removing, the student learns all about the construction and operation of the machine tool and the cutting tool, including the limits of speed and feed and a study of the effects of different materials on cutting tools, the effects of varying shapes and angles of cutting tools, etc. The machine shop should be a laboratory of research and not a manual training school when it is made part of the work of technical colleges of university grade. With properly educated and properly paid instructors in charge of the shop-work, the university machine shop can be made a research laboratory. At the same time it can well be, without any stigma whatever, a training school for those who will go into manufacturing work.

A student need be none the less prepared with general principles if his shop-work has given him some idea of the proper times in which work should be done, or the best types of modern tools with which work should be done. His work in machine design can be just as effective if it includes the harmonizing of design with economical manufacturing processes.

And in order that the student may get this kind of training, his instructor must know commercial practice. The best law schools and medical schools would not think of employing instructors who had not had experience as practitioners. The director of a university machine shop should be a man with the capacity and receiving the salary of a good works manager. There will be no danger that a broad-minded man with such capacity would convert his department into a sort of elementary business college, as is so often feared by pedagogues.

An experienced practitioner would realize the limitations of students and would know what elementary principles needed emphasizing. He would lay more stress on elaborate imitation of so-called standard commercial tests than is apt to be laid on them by the non-practitioner. The experienced practitioner would care little whether his laboratory courses conformed in every detail to those given in some other school. If he were the right sort he would blaze his own trail.

A recent contributor to an engineering publication told of how he was conducted through the "shops" of an engineering school, where he saw boys at lathes "nibbling" off bits of metal from nameless and aimless iron castings, and the head of the department gave very vague and evasive replies to the visitor's pertinent and practical questions. The contributor's indictment is not fair to make in a general way, since it does not apply at all to our high grade engineering schools. Among the colleges and universities that cannot be placed in the first rank, however, and particularly among those located in small towns removed from the living accomplishments and contact of engineers, we may still find many examples of the old-time practice of putting a professor who has originally taught mathematics or "pure" science in charge of engineering. Where such selections are made because of the financial inability to engage men with broad fundamental training and actual engineering experience, it would have been better not to attempt to establish an engineering department. There is, however, another reason for such appointments, namely, an echo of the Oxford spirit, which cannot conceive of practical engineers having broad interests. In colleges of such isolated locations one finds professional men of excellent accomplishments in their own line, who have a deep-seated social prejudice against any delving into the actualities of manufacture, and this prejudice takes form in a belief that an engineer is not sufficiently appreciative of the importance of fundamental principles. Several of these good men have expressed surprise that I should find pleasure in my moments of relaxation in reading what they themselves considered good non-technical literature, and I have always felt that they had a suspicion that I held the book upside down when I read it. The academic atmosphere and artificial view of life permeating universities is creative of the sentiment that makes important the preservation of those fields of investigation which do not enter into everyday life. This atmosphere and spirit will account for university authorities considering it more desirable to have full professorships of Greek History and Cryptogamic Botany than of engineering. An investigation of State university and college catalogues reveals that, whereas the colleges of arts and the schools of medicine and law are supplied with an abundance of full professorships, the technical branches of engineering are, in many prosperous schools, assigned to men ranking as instructors and assistant or associate professors, although acting as heads of

their respective departments. This in face of the fact that in these same schools the number of students taking engineering outnumber those of any other courses. One reason for this failure to recognize the needs of engineering is the fact that engineers do not take the part in public life that is taken by lawyers, physicians and farmers. The "Grange" and Agricultural Society keeps in close touch with the State Agricultural College. While the farmer keeps his eye on his representative, the busy city engineer seldom seeks to interest lawmakers in providing for the engineering departments of these agricultural colleges, which have great possibilities as American polytechnic schools in the true sense of the word. Similarly, when appropriations and distributions are made for State universities, we find Bar Associations, Medical Societies and Editorial Conventions taking active part in seeing that their charges are properly fostered, while the engineers are never heard from.

For the good of the community, as well as the good of the profession and the rising generation, the engineer must become more of a factor in public life, and this very subject under discussion should be one of interest to practicing engineers and be given more thought and attention than it has been receiving from them.

Prof. Arthur L. Williston.—I think the subject of this paper is one which is of interest, not alone to the teaching members of our fraternity, but to all the members of the Society for the experimental laboratory work, better perhaps than any other part of the courses planned for the education of engineers, may be made to give to the student a clear and definite idea of the true character of engineering and the kind of problems that the modern engineer meets daily in his work—and we are all interested in having the young men who are about to take up their professions start in on their life work with a right conception of its nature. I, therefore, want to express my appreciation of the clear and forceful way in which the writer of this paper has pointed out to us some of the very grave changes of certain methods of conducting such work; and has held up an ideal toward which all who have charge of experimental laboratories should strive.

It seems to me that the type of work which the author so justly characterizes as the *"Laboratory report blank," "Commercial Test," cut* and *dried* kind of laboratory instruction—which is, unfortunately, still so common—has no place in a progressive engineering school to-day. Conditions have changed during the

last twenty or thirty years and the field of mechanical engineering has wonderfully expanded. The making of *efficiency tests* constitutes but a very small, almost insignificant, part of the engineering work of the present time. By far the largest part of the graduates of the engineering schools enter manufacturing in one capacity or another, and the demand is for men, as Dr. Lucke says, who are "trained to do and to think," and not for men who are drilled in the proper form in which to make efficiency test reports. The students in the laboratory should rather be impressed with the almost infinite number of things that they need to find out about the various pieces of apparatus.

In his main thought I am in hearty sympathy with the author, and in substance I believe I am all the way through. In one particular, however, I think he has been somewhat unfortunate. The term used to describe what he considers the better laboratory method—*research*—is likely to be misleading. The other term which he uses synonymously with it later in the paper, "Successive Problem Solution," expresses the idea of small pieces of work which will give training in both doing and thinking and which may be simple enough for undergraduate students. This is more suitable than the term research, which is apt to be associated with advanced and somewhat elaborate scientific investigations. Also the author gives the impression that a more extensive equipment is needed for the type of work that he advocates than is required for the other kind of work. To this I cannot agree. Plenty of equipment is of course desirable, but the "problem" method is superior to the "test" method with the most meager equipment. Even though there is nothing but a worn-out slide valve engine, the problem of finding out how much conditions will be improved if the valve is carefully scraped to a bearing on its seat will prove of more value than an efficiency test on the same engine.

Prof. D. S. Jacobus.—I am heartily in sympathy with the ideas brought out by the author of the paper, and I wish to congratulate Professor Hutton for coining such a fitting expression as "the cook book method."* For a number of years I have contended that there can be too complete a system of working notes and blanks in a laboratory course. If a system of laboratory blanks are very

* Dr. Lucke's paper was presented by Prof. Hutton, who said that he has designated that system of instruction where the students can follow notes so closely in their laboratory course that they do not have to think for themselves as "the cook book method."

complete they relieve the instructors of much work, but the students may not derive as much good from the course as they would with less complete blanks and more individual instruction. Of course there may be too little system in the notes as well as too much, but of the two evils I would rather see too little than too much.

For a number of years instruction in the experimental field has been my specialty—first as assistant to Professor Denton, who was the originator of a systematic course of such work in connection with mechanical engineering instruction, and later as head of the Carnegie Laboratory of Engineering at Stevens Institute. I have tried a number of different methods and watched the results. We have had too much system as well as too little; in fact the printed report blank given as a sample of the extreme to which such a system may be carried is similar to a blank, now discarded, which was used in our work. An incorrect conception of such blanks may, however, be obtained if the entire collection is not examined and the conditions under which they are employed are not known. Naturally at first we started with no particular system, and a great many of the older graduates now come back to us and say, "Well, your fine equipment is all very nice, but I do not think the students get as much out of it as we did in the old days when we worked in the dark basement of the main building." At that time nearly all of the instruction was given individually, and but few blanks were used, so that in working out the results the students had necessarily to become acquainted with the methods before making any computations. On the other hand, I feel that the boys of to-day will probably say much the same thing to the men of some other classes later on, for there is always a certain romance in looking back on the work done in our younger days, and this may make what appears ordinary to us at the time seem of much greater value as we look back on it in the light of later experience.

Now let me tell you what has been built up in the way of a course at the Stevens Institute. We believe in system, but, as I have said, not too much of it. We now start a class on work where they are made to follow a rigid system of instruction. In the lecture room we give demonstrations with full-size machines and apparatus. For example, if we are going to test steam-engine indicators, the class meets for a lecture, where we explain the method, tell why it is the best method and how errors may be in-

volved by using other methods. The students then go to the laboratory and make tests of indicators, handling the apparatus themselves and reporting the results.

After the students go through a preliminary training in this sort of work they are assigned different problems on various machines. Here a mistake may be made in making the problems too difficult. The boys will not remember all they do, and by repeating certain steps where they are thrown upon their own responsibility, they are bound to find that there is something which they did not appreciate at first. So we assign to the students comparatively simple problems and allow them to work them out by themselves, and require them to present a report on the methods employed and the results obtained. The reports are scrutinized very carefully by the instructors, and we aim to make the students present a report in good English of the general type they should later on, if required to present a report on the results of their work, to a concern by which they are employed.

I think this method is the best one—that is, first to give the principles in a systematic course, and then throw the students on their own responsibility in applying them.

The problems assigned, where the students are thrown upon their own responsibility, are not in the nature of research work, and in the regular part of the course cannot be made of this class, because where the men are learning they are bound to make mistakes, and by the time they would become sufficiently drilled in securing exact data for some special work there would be no more time available for completing it. This applies to the regular course in experimental work, and not to that part of the course assigned near the end of the senior year to thesis work, where problems in research work may be undertaken by the students should they so desire. In the regular course of experimental work we simply assign the students special tests, and when they are left to their own responsibility we find that they are apt to make a great many mistakes, and in correcting these mistakes they gain a corresponding amount of information.

There is one feature which I think is particularly important in this work. This is the examination of the note books in which the original test data are entered. At one time we furnished data blanks to be filled out, but now we make the students use a standard form of note book in which they write out their own headings and rule up tables for the data in the same way they would in

practical work. As soon as a group of students complete a test, and before leaving the ground, we require each man to work out the principal results without the use of any notes or instructions other than those given verbally by the instructor. No very great refinement is required in this work, because ordinarily there is not enough time available, but the men are required to work out the principal results with reasonable accuracy directly in the note books containing the original data, and they are made to understand the methods employed. These original notes are examined by the instructors so that each man's work is carefully followed.

After having worked out the principal results in the original note books, the students report in the computation room and prepare a final report. Here we make use of the forms given in the reports of various committees of this Society. We have in some cases reproduced the blank tables, and also have bound copies of the complete reports for reference. The blank tables are used for some of the tests, whereas in others the students prepare their own tables.

To sum up the whole matter, I believe in system in this sort of work, but not too much of it, and I think that phrase coined by Professor Hutton, " the cook book method," expresses very well what I have contended is the wrong way to teach experimental work in an engineering school.

Mr. H. H. Suplee.—I had the opportunity of discussing this subject two or three years ago with Professor Unwin, in London, and he called attention to the fact that what Professor Hutton has called " the cook book method " can be carried too far. He expressed himself as rather opposed to the use of too many forms and codes in actual engineering practice in testing, partly because it enabled a man who was really not competent to fill out what looked like a respectable report without doing much real work, and also because a competent engineer was often turned away from discussing original phenomena which he might otherwise make the leading features of his report. He believed that we were codifying and cataloguing our methods of engineering education too much to get the best results.

Dr. Lucke speaks of the criticisms of some of our foreign visitors, but I think these were practically all Englishmen, and I wish to say a word about the continental engineering laboratories. I think some of them are quite as well equipped as those we have, and the original research method there seems to be carried further

than in this country. Any one will see this in the fine laboratory of Professor Stodola, at Zürich, and in the great laboratories at Charlottenburg, as well as in the laboratory of Professor Schröter, at Munich. The professors there spend a part of their time in original research, investigating scientific and engineering phenomena which really need to be found out for the benefit of the profession, and they use their students as assistants. It seems to me that it is a waste of some of our very magnificent equipments in our laboratories if they are not used in making original contributions to the work of the profession.

*Dr. Lucke.**—The discussion of this paper by the members of the Society has brought out the very gratifying fact that the methods advocated in the paper are not only appreciated as proper ones, but are actually in use in more than one institution. The broad general principle may of course be varied in detail almost infinitely without detracting from the value of the instruction. Such variations are necessary to meet the needs of the different institutions. It seems probable that the old method, referred to by Professor Hutton as the "cook book method," will before long be abandoned in all of the first-class schools of technology, and in its place will be found instruction leading to "how to do things," rather than instruction which seeks to fix in the memory a large number of facts.

*** Author's Closure, under the Rules.**

No. 1088.*

THE MICROSTRUCTURE AND FRICTIONAL CHARACTERISTICS IN BEARING METALS.

BY MELVIN PRICE, LINCOLN, NEB.
(Associate Member of the Society.)

1. The investigation here recorded was carried out in the laboratories of the department of Mechanical Engineering and the department of Metallurgy at Columbia University. The main questions considered were: (1) The law of variation of the amount of friction with the velocity of sliding, other conditions remaining constant. (2) The theory of Charpy and others regarding the function of the hard grains and of the plastic ground mass in the microstructure of alloys. (3) The theory of Professor Goodman, which received the credence of Prof. W. C. Roberts-Austen, the eminent metallurgist, which theory related to the effect produced by elements of high atomic volume as distinguished from the effect of those of low atomic volume in bearing alloys.

2. Regarding the first of these questions, in the earlier years of the last century, Morin made many observations which he interpreted to mean that the force of friction is independent of velocity. Many later investigators have made determinations which they interpret to mean that Morin's conclusion could not have been correct, and some have attempted to replace Morin's law of independence of velocity by a law in which the whole range of ordinary working speeds is divided into three or more gradations, each of which has a separate characteristic regarding speed-friction variation.

3. The theory of Charpy evolved through an investigation of the metallurgy of a number of alloys belonging to the types commonly noticed in use in machine practice for bearings was

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

that the friction of hard substances was less than that of soft ones, and that, therefore, the hard grains in alloys supported the load and contributed to the alloy whatever anti-friction quality it possessed, while the plastic ground mass in which the hard crystals were embedded served to allow the bearing step to mould itself around the shaft, producing a better fit. It is clear from his account that his conclusion is not based upon frictional tests. He attempted to differentiate between the metals from this standpoint by using the apparatus devised by Le Chatelier. But after many trials they concluded that with the apparatus at hand absolutely no difference could be detected between the resistances of alloys when lubricated, even when such alloys differed widely as to physical properties and microstructure, and that there was no certain difference when run in the dry state. Charpy then proceeded to examine a number of industrial bearing alloys from a metallurgical standpoint, supplementing the investigation by a number of compressive tests. The impression left with the reader of his paper is that he believed himself able to justify the use of these industrial alloys (which had been empirically developed) by assuming that the hard isolated crystals of chemically combined metals found in the microstructure, were the factors essential for the support of the load, and that being hard substances their friction was less than that of the soft matrix, the plastic property of which matrix as exhibited in his compressive tests completed the justification for the use of such alloys.

4. Goodman's announcement was to the effect that whenever a bearing alloy supposed nominally to contain the ordinary *pure* metals commonly used for such purposes had added to it a third constituent, the friction was raised or lowered in value according as the added constituent was of low or high atomic volume. In a discussion at a meeting of the Institution of Mechanical Engineers, April, 1895, Professor Goodman is quoted as having early noticed great differences in the resistance of alloys which in tentative analyses had shown practically identical compositions. Further and more complete analyses, however, showed minute quantities of impurities. Going deeper it was found that when these impurities were elements of low atomic volume the resistance was increased, but when of high atomic volume the reverse kind of performance was observed. After much experimenting, he generalized by establishing as a "law" that a good bearing metal must have at least one constituent of high atomic volume, or in words, regard-

ing his investigation, "it showed indeed clearly the importance of getting absolute purity in metals used for antifriction alloys, that is purity at least in the sense of keeping out metals of small atomic volume."

5. Laws of friction.—The laws of friction as announced by Morin were as follows:

(1) The friction between two bodies is directly proportioned to the pressure, *i.e.*, the coefficient is constant for all pressures.

(2) The coefficient and amount of friction, pressure being the same, is independent of the areas in contact.

(3) The coefficient of friction is independent of velocity, although static friction (friction of rest) is greater than friction of motion.

6. In direct contrast with the above, we find the modern beliefs regarding frictional behavior stated by Mr. John Goodman in a review of the results obtained from the testing machines of Thurston, Tower, and Stroudley, as follows:

(1) The coefficient of friction for moderate pressures and speeds varies approximately inversely, as the normal pressure.

(2) The frictional resistance varies as the area in contact, the normal pressure remaining constant.

(3) At very low journal speeds the coefficient of friction is abnormally high; but as the speed of sliding increases from about 10 to 100 ft. per min. the friction diminishes, and again rises when that speed is exceeded, varying approximately as the square root of the speed.

7. In support of the former laws, deduced by M. Morin, there is the work of Coulomb and that of Rennie, and of others less notable. Morin's work itself is so extensive as to be pretty convincing, his three *Mémoires* containing no less than 178 series of not a few cases each, the experiments having been performed under varying conditions of material, area of contact, pressure, velocity, rest, retarded movement, accelerated movement, uniform movement, lubrication, etc. The apparatus by means of which Morin's laws were obtained consisted essentially of a *traineau*, or sled, whose performance upon a rectilinear track was studied.

Latterly we have seen for the friction test various contrivances, nearly all of which differ radically from the ingenious mode of Morin. Rotative devices are now the most commonly used. In these the moment of the friction is usually measured directly. By means of a rather good machine of the latter type Mr. Beau-

champ Tower has given to us the results which perhaps form the best basis for a law of friction of which we now have knowledge. These seem to establish a relation between the force of friction and the area in contact. The following figures, due to John Goodman, from Tower's results, were taken from Kent's Pocket-book:

Load in pounds per sq. in.....	529	468	415	310	258	205	153	100
Frictional resistance per sq. in.....	.416	.498	.472	.464	.438	.43	.458	.45

Thus if we let

A = area in contact.
P = total normal pressure.
F = total resistance to sliding.
p = normal stress.
$\frac{F}{A}$ = constant, although p is a variable.

8. Recognizing the obvious fact that no two surfaces when placed into contact can be conceived to fit exactly except in case the normal pressure is made great enough to overcome the tendency toward *point*, or small area contact, and calling the total area of the smaller surface the apparent area, and the summation of all small areas actually touching the real area, we may adopt the following further symbols:

A = apparent area in contact.
A' = real area in contact.
p = apparent stress over area.
p' = real stress over area.
μ = coefficient of friction.

9. Now suppose a uniformly distributed load be applied to the upper body: then probably the elastic material of this body will deflect so as to place greater area into coincidence. A question which cannot be answered with certainty because of incomplete knowledge of initial internal stresses, unequal rigidities, etc., is as to *how the real area of contact varies with the total pressure on a body*. But, making the simplest assumption consistent with facts, let us say that A' varies directly as P. Now $p'A'=P$, so that the above assumption leads to making p', the mean stress over the actual tangent area, a constant quantity. From the

fundamental definition we have, $F = \mu P$, and dividing by A' we get

$$\frac{F}{A'} = \mu \frac{P}{A'} = \mu p',$$

in which p' is a constant quantity by our hypothesis.

10. It is probably quite reasonable to assume that in the experiments of Tower the real areas were very closely equal to the apparent ones, since the above quoted results agree very well and, in fact, everywhere his work is consistent. But he obtained $\frac{F}{A}$ = a constant. If A in his case he put $= A'$, then his results may be applied to our present question, and we may put $\frac{F}{A'}$ = constant. So that if we keep increasing the normal pressure P from a small value upward, A' will increase proportionally, by hypothesis, until it approaches very nearly the apparent area A. But below the latter critical value we have p' = constant, and $\frac{F}{A'}$ = constant, the former by hypothesis and the latter by Tower. In the expression

$$\frac{F}{A'} = \mu p'$$

we must then have

$$\mu = \text{constant}$$

up to the critical point where $A' = A$, although we have been *varying* the applied pressure P. If $\mu =$ constant within this range, remembering that $F = \mu P$, we have the *friction varying directly as the pressure* P within the range of varying contact, provided Tower's conclusions are correct in experiments conceived to have been carried out wholly above and outside that range. If $p = \frac{P}{A}$ = the apparent stress it is seen, since $P = pA = p'A'$, A' is always less than A, and consequently p' always greater than p except at and above the critical point where $A' = A$ and $p' = p$.

11. It is thus seen that there may be a well-defined condition where the force of friction varies directly as the total pressure; with a constant apparent area, directly also as the apparent stress. But this is the first law of Morin.

12. The latter's law regarding contact areas is more difficult to

account for. If, for instance, we could write, when the apparent surface was arbitrarily varied,

$$A' = K_1 A$$

where K_1 = a constant. Then still adhering to $\frac{F}{A'}$ = a constant, and reckoning that P = a constant, we should have

$$F = \mu P,$$

$$\frac{F}{A'} = \frac{\mu P}{A'} = \text{constant} = K_2,$$

$$\mu = \frac{K_2 A'}{P} = \frac{K_1 K_2 A}{P} = K_3 A,$$

or μ varies as the apparent area of contact surface where Morin often repeated "*Independant de etendue de la surface de contact.*" But we must remember that we are still under the hypothesis applying to low pressures, below the critical point, at which real contact area becomes equal to apparent contact area, and that under such conditions the contact is very likely to be that of points rather than areas. Now since three points of such contact may stably support a body on a surface, it is in such cases extremely unlikely that $A' = KA$, as it was above assumed. Rather it is probable, when the apparent surface was arbitrarily varied in Morin's experiments, a new set of three or more supporting points came into play, the small surrounding areas of which were not sensibly different from the corresponding areas formerly, under which circumstances the apparent area of contact would of course make no difference in the resistance.

13. To accept the above explanation of the reasons for Morin's results as they bear upon the first and second laws of friction, it would be essential to assume that Morin's contact surfaces did not fit well. Whatever the probabilities, we may very well continue in this assumption when we come to examine the third law. In this connection the following experiment is submitted: A suitable shaft was loaded at its middle by mounting thereon a brass disc, and at each end it was properly journalled to run in half-bearings made of pure copper. Provision was made for lifting this rotating system clear of the copper bearings into contact with a moving belt, which imparted to the system a rapid motion of rotation. When the speed of rotation became as great as desired the shaft was lowered into contact with the copper half-bearings and the lift disengaged, while the system continued its rotation freely

except for the friction of the journals. Its performance was then noted carefully until it came to rest. By means of an instrument hereinafter described it was possible to observe the events at which the disc and shaft came to certain known speeds. And by counting the times between these events, a retardation record was possible. In the following table are given values of times in seconds during which the speed fell from a certain number of revolutions per second to a speed of five revolutions per second slower. The figures at the headline designated by N are the numbers of the periods between events, reckoning the first period as that between the event of rest and the event at which the speed was five rev. per sec., etc.

W means cleaned and lubricated.

U means lubricated.

Each horizontal line is a retardation record, and the records were performed in the order in which they are set down in the table.

14. Undeniably, in the first portion of the table, the lengths of the periods—and therefore the frictional resistances—are nearly constant for all speeds from the high one at starting down to very near the point of rest. Undeniably, also, in the latter portion, there is a persistent lengthening of the periods 2, 3, 4, and 5, making the friction seem to depend upon the speed. The corresponding change in the conditions was manifestly one of mechanical fit. Toward the end of the experiment the journal became more and more perfectly "worn to" the bearing.

15. This probably shows, with what we have shown above, that Morin's laws may be explained as being those obeyed by bodies in contact only at small point areas whereat they mesh deeply: that the modern laws are obeyed by bodies in better mechanical contact, but meshing less deeply.

16. So far it has been shown that, contrary to the announcement made by M. Morin, the resistance may not always be independent of velocity. We shall now turn to an investigation carried forward for the purpose of finding out whether or not there is a definite law of dependence, such as the one quoted above, or whether the variation appears as characteristic of the given material.

17. A number of alloys were made and ingots cast into bars of suitable shape, from which sections were cut, polished and examined with the microscope. Suitable half-bearings were then

prepared for the testing machine, and from these the frictional retardation curves were obtained. The following method was used for observing the manner in which the friction varied with the speed:

18. From the consideration of some of the difficulties with the

N.....	14	13	12	11	10	9	8	7	6	5	4	3	2	1	
.....	....	....	....	....	....	6	6	6	6	5	7	6	7	7	.....W
.....	....	....	8	7	8	8	7.5	8	7.5	8	8	8	9	8	.....W
.....	....	....	....	....	....	....	....	7	7.5	7.5	8	7	7.5	6.5	.....W
.....	....	....	....	....	7	8	9	9	8	9	8.5	8	8.5	7	.....W
.....	....	....	....	....	7.5	8.5	8	8.5	7.5	8.5	8.5	8.5	8.5	7.5	.....W
.....	....	....	....	....	....	....	8	7	7	8	8	9	8	8	.....W
.....	....	8	8	8.5	8.5	8.5	8.5	9	9	9.5	6	13.5	9	8	.....U
.....	....	....	....	....	....	....	10.5	10.5	9.5	10	10	10	9.5	8	.....W
.....	....	....	....	....	9	8.5	8.5	9	9	9.5	9	9.5	10	9	.....U
.....	....	....	....	....	....	10	9.5	9.5	10	9.5	10	10	10	8	.....W
.....	....	9	8.5	8.5	8.5	8.5	8.5	8.5	9	9	9	10	8	7	.....U
.....	....	....	....	9	9	8	8	8.5	9.5	9.5	9	9.5	9.5	8.5	.....W
.....	....	....	....	....	....	....	8.5	8	8.5	9.5	10	10	10.5	9.5	.....U
.....	....	....	8	8	8	8.5	7.5	8.5	8	9	9.5	9	9.5	7	.....U
.....	....	....	....	....	11	10	11	10	9.5	10	10.5	10	10	8	.....W
.....	....	....	....	8	8	8.5	7.5	8	8	8	10	9.5	10.5	8	.....U
.....	....	....	....	8	8	8	9.5	8.5	10	10	10	11	11.5	9.5	.....W
.....	....	....	10.5	9.5	9	9.5	9.5	11	11	12	14	14	13	9	.....U
.....	....	....	....	....	....	10	10	10	10	10	10	13	13	9	.....
.....	....	....	....	9	10	9	9	11	10	11	10	12	12	10	.....W
.....	....	....	....	....	12	11	11	10.5	9.5	10	12	12	13	9	.....U
.....	....	....	....	....	10	11	11	11	10.5	10.5	10.5	12.5	12	9.5	.....
.....	....	....	....	....	7	9	9	9	10.5	11.5	11	13	10	9	.....W
.....	....	....	....	....	....	9.5	9.5	10	12.5	11.5	11	13	9	8	.....U
.....	....	....	....	....	....	8	6	7	8	8	8.5	8.5	10.5	7.5	.....
.....	....	....	....	....	10	14	17	9	13	15	14	12	11	9	.....W
.....	....	....	....	....	....	....	....	15	14	14	14	14	13	9	.....U
.....	....	....	....	....	10	12	12	12	14	15	14	15	13	9	.....
.....	....	....	....	....	....	14	13.5	13.5	14	15.5	16	14.5	13.5	10	.....W
.....	....	....	....	....	....	12	11	12	14	13	13	13	12	10	.....U
.....	....	....	....	....	....	....	10	10	11	13	13	13	11	10.5	.....
.....	....	....	....	....	....	....	12	13	13	15	15	14	13	10	.....W
.....	....	....	....	....	....	....	11	11	13	11	15	16	13	12	.....U
.....	....	....	....	....	....	....	....	12	11	14	12	13	12	12	.....
.....	....	....	....	7	8	8	7	8	9	9	9	8	9	6	.....W
.....	....	....	....	9	9.5	9.5	9.5	10.5	11	11	10	9.5	8.5	6	.....U
.....	....	8.5	8	9	9	7.5	7	7	9	8.5	7	7	7	7	.....
.....	....	....	9	8	8	9	9	9	9.5	9.5	9	10	9	6.5	.....W
.....	....	....	9.5	9.5	8.5	8.5	7.5	8	9	10	10	10	9	8	.....U
.....	....	....	7	6	7	7	7.5	8.5	8.5	8.5	9.5	10	9.5	6	.....
.....	....	....	....	9	9	9.5	9.5	10.5	9.5	10	9.5	10.5	9	7	.....W
.....	9	10	9.5	10	9.5	10	10.5	10.5	10.5	10.5	10	10.5	10.5	7.5	.....U
.....	....	....	6	6	6	6	6	6.5	7	6.5	7	7	6	7.5	.....
.....	8	9	9.5	10.5	8	9	9	8	7.5	8.5	8	9	8	7.5	.....W
.....	....	8	7.5	7.5	7	6	7	8	10	11	11.5	10.5	10	7	.....U
.....	....	....	7	7	6.5	7	7	8	7.5	8	8	8	8	7	.....
.....	....	....	....	....	13	13.5	12.5	13.5	13.5	14	14	14	13	12	.....W
.....	11.5	12.5	12	13	13.5	14.5	15	16.5	16.5	17	17	15	15	13	.....U
.....	....	10	8	8	9	9	9	9	9	9	10	11	12	11	.....
.....	....	12	11	14	15	15	16.5	16.5	19	19	17	18	18	13	.....W
.....	10	11	10	11	9	11	12	13	14	16	16.5	18.5	17	13	.....U
.....	....	....	....	....	....	....	14	17	16	16	17	17	16	11	.....W
.....	10	11	10	11	10	10	11	11	13	15	16	17.5	17.5	12	.....U
.....	....	10	9	10	10	2	0	12	13	16	17	11	19	13	.....W
.....	....	7	8	10	8	9	8	10	9	9	10	10	10	10	.....U
.....	12	10	9	8	8	10	12	9	10	10.5	10.5	11	15	13	.....W
.....	....	8.5	8.5	9	9	10	9	9	8	9	17	19	18	11	.....U
.....	....	9	9.5	11.5	10	10	9	11	11	12.5	15	18.5	19	13	.....W
.....	....	12	14	13.5	12.5	12	11	14	14	16	21	30	17	15	.....U
.....	11	13	11	12	11	12.5	13	13.5	14	14	16.5	18.5	18	15	.....W
.....	11	11	11	13	13	13	13.5	15.5	16	20	21	23	21	15	.....U
.....	....	....	13	11	11	12	13	15	16	17	20	21.5	20.5	14	.....W
.....	12	14	11	11	11.5	14.5	15	17	20	21	23	29	19	16	.....U
.....	....	9.5	10	12	12	12	13	13	15	17	21.5	18	17	15	.....W
.....	12	12	11	12	13	14.5	16	17.5	21	25	27	28	28.5	18.5	.....U
.....	10	11	11	10	10	11	14.5	16	16	18	21	23.5	27.5	31	21	.....W

methods of Thurston, Tower, Denton, Wellington, and Chatelier, and of the great differences noted in the results from them, considered as absolute measurements, together with what was noticed in the microscopic examinations of bearings which had been some time in operation, the writer was led to seek a solution of the questions of friction through other means. That which had been observed by others, indeed, that which might have been accepted as the truth merely by process of deduction from probability, was confirmed upon examination of a large "white-metal" bearing which had been some time in operation in an ordinary shop situation. In this case the original longitudinal markings made in fitting the journal were visible over the greater portion of the cylindrical surface of the white-metal step. Indeed the very smooth portions, together with those where abrasion had occurred, indicating places where the journal had actually touched the bearing, in metallic contact, comprised an area not greater than 2 per cent. of the whole surface. Of this small percentage a large portion was plainly due to the abnormal presence of dust or grit, these places being near the ends: so that the really smooth-worn area, situated further inward, was a very small fraction of the apparent area in contact. To be fair, it should be said that a second, smaller bearing examined exhibited indications of a much greater area as having been touched by the journal. But in this latter case the markings showed great furrows, due to abrasion and to the further fact that the journal was very untrue, exhibiting irregularities, errors of machining. It was not thought that this latter investigation controverted the obvious fact deducible from the former, viz., in practice, and therefore in machines designed to measure the force of friction, which imitate practical conditions, great uncertainty must exist as to the *real* area of the contact surface. If, then, any law of friction depended upon this area, its truth or falsity could hardly be demonstrated by means of an ordinary practical bearing.

19. It occurred to the writer to attempt to measure the force of friction by means of the retardation it produced in the speed of a rotating shaft loaded by having mounted concentrically upon it a heavy disc, when the rotation was subjected to no appreciable retarding or other force except the friction of the two journals, at the shaft ends, in simple half (or less) bearing steps (see Fig. 239). For observing the fall in speed an instrument due to Mr. F. H. Ball, Jr., which may be called the vibrating tachometer (Fig. 240),

was employed. The latter consists essentially of a long stiff steel bar, fixed at one end, horizontally, and made to tremble by means of an instantaneous contact electro-magnetic arrangement similar to that employed in electric bells. At the free end is attached to the bar, in a vertical plane, a screen containing a narrow horizontal slot. When stationary, this slot just opposes an exactly similar one cut in a stationary screen fixed in the vertical plane, parallel and

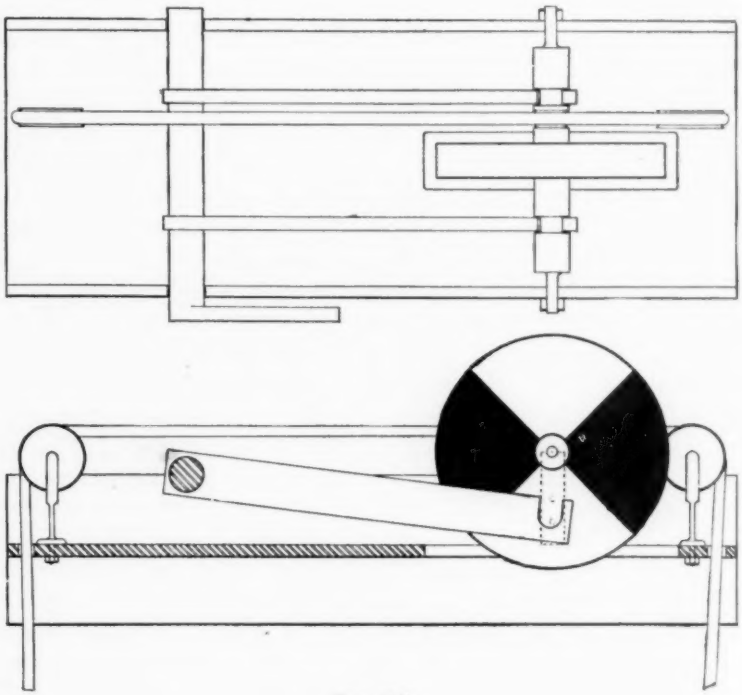


FIG. 239.

very near to the screen carried by the vibrating bar. When the movable screen is set vibrating at a sufficiently high frequency any recurring or cyclic motion observed through the stationary screen will exhibit at successive distinct instants of time the particular phases of its action which are occurring when the vibrating and stationary screens are opposed. It follows that if the frequency of the vibration is equal to that of the cycle observed, the summation of all the effects upon the eye will give a vision of the system whose cycle is being observed, which is that of the system when

stationary. The same sort of vision will be observed when the recurrence of the system has a frequency any whole number of times as great as that of the vibrating screen, within the power of the eye to recognize form, stationary, produced by successive instantaneous images at some interval of time apart when disturbed by visions of the same form in motion during those intervals. Also, in case the cycle observed is symmetrical and certain portions superposable to fit upon others, then, when the system recurs at other frequencies than any of those mentioned above, similar images will be visible, due to instantaneous superposition of like phases. Lastly, another stationary effect will be noticeable, in the latter case, when certain phases are *partly superposed,*

FIG. 240.

recognizable from some peculiarity of figure due to the instantaneously superposed images. As applied to the case in hand, by specially marking the rotating disc, and noting the times between the recurrences of stationary images, while the motion was retarded only by friction, from a knowledge of the frequency of vibration of the screen, the retardation of the motion of the rotating system could be computed. When these times were plotted against speeds as ordinates, the resulting curve furnished a graphical record of the friction throughout the performance. Let

φ = friction angle.
F = the force of friction at periphery of journal.
W = weight of the rotating shaft and disc.
K = radius of gyration of the rotating mass.
r = radius, in feet, of the journal.
a = linear retardation in ft. sec.^{2}
n = fall of speed in r.p.m. in time t.

Then $$a = \frac{2\pi n K}{60t} \text{ and } F = W \sin \varphi.$$

The equality of moments is

$$\left(\frac{a W}{g}\right) K = Fr,$$

$$\frac{F}{W} = \frac{a}{g} \times \frac{K}{r} = \sin \varphi = \frac{\pi K^2 n}{30grt}$$

$$\tan \varphi = \mu = \text{coefficient of friction.}$$

But $\frac{n}{t}$ = slope of the retardation curve = $\tan \alpha$.

The friction angle is so small that we may say

$$\sin \varphi = \tan \varphi = \mu$$

very nearly. Making these substitutions, we have

$$\mu = \frac{\pi K^2}{30gr} \tan \alpha,$$

or the coefficient of friction is at every point very nearly proportional to the slope of the curve.

20. Two bearing steps were fastened to the sides of a steel frame supported at a convenient height. The rotating shaft was $\frac{3}{4}$ inch in diameter at the middle and was cut down to $\frac{1}{4}$ inch in diameter at each end where the steps were fitted. In order to reduce to a minimum the liability of having large apparent areas of contact, but small real ones, these bearing steps were cut to a longitudinal width of about 0.04 inch, while their circumferential arc embraced considerably less than half of the $\frac{1}{4}$-inch journal. The journal was slightly scored to permit only a small amount of axial motion. At the center of the shaft was mounted a solid brass disc $\frac{5}{8}$ inch thick and 5 inches in outside diameter. The weight of the disc and shaft was 4.14 pounds. At a short distance away on each side of the disc a journal $\frac{1}{2}$ inch in diameter was cut, which ran in bearings at the ends of two arms which were pivoted at a short distance away to the frame of the machine. Attached to these a handle was provided, by means of which the arms could be raised, hooking their ends under the shaft in the journals provided, thus enabling the observer to raise the rotating system clear of the test bearings into contact with a moving belt, and when at speed to set it down again upon the test bearings and, unhooking the arms, to allow the system to spin

freely, except for the friction, when its performance could be observed by the tachometer, suitably arranged a few feet away. The source of power was a small electric motor, provided with a switch for stopping it quickly at will; a round belt from the motor's pulley traveled over a guide pulley at each end of the frame above mentioned. This belt ran in a horizontal direction an inch or so above the shaft, so that when motion was to be imparted to the rotating mass, it was only necessary to hook the arms under it and raise it clear of the test bearings until a circumferential groove in the shaft engaged with the belt, setting the system into rotation.

21. Alternate quadrants of the brass disc were blackened and the remaining two whitened. Then obviously what may be called a velocity "node" would occur when its speed was such that it made one-half a revolution in the same time in which the screen made one-half oscillation; also when it made a whole revolution per half-oscillation; and also when it went one and one-half revolutions, etc., or nodes would occur at shaft speeds of 1, 2, 3, 4—n times the frequency of oscillation. It was also possible with a little practice to observe the nodes at greater speeds, these being distinguishable from a certain appearance of uniformly gray color on the disc, apparently stationary, as in the case of the half-nodes. These occured when the disc revolved one-fourth a circumference in the same time that the screen made a half-oscillation.

22. Thus it was found possible to observe the change in speed of the disc after setting it free on the test bearings by observation of the times between the "node" events, which occurred at speeds which could be reckoned as some simple multiple of one-half of the frequency of vibration of the screen. With the favorite arrangement, for example, the frequency of oscillation of the bar was about 10 per second: so that nodes occurred at disc-speeds of

5, 10, 15, 20 ..... $\frac{10n}{2}$ revolutions per second.

23. At first it appeared that since great uncertainty must exist as to the amount of area really in metallic contact in large bearings, it would be well to eliminate as much of this uncertainty as possible by reducing the bearing area. Be it well or ill, it has been assumed that with smaller areas to attend to, very probably the *apparent* area would represent more nearly the *true* area in contact. Hence it was decided to work with small areas. But this implied small normal forces and also very light resistances. Diffi-

culty loomed up when an attempt was made to conceive apparatus whose own resistance would be inconsiderable enough not to effect very adversely any result for the friction itself. In the method adopted the observing apparatus was entirely detached from the system being observed, as was every other disturbing influence except the atmospheric friction, and whatever vibrations were set up by the spinning system itself.

24. Another part of the work lay in the field of metallurgy. In that connection a number of alloys were made, in which the major number of the different structures which had been noticed might be expected to exist. A number of these were not unlike empirical bearing metals of commerce. These were utilized in the present investigation in the following manner: To a bearing alloy, which usually was very rich in some one of the softer metals, as lead, tin, aluminum, copper, etc., a third constituent would be added in three different proportions. The plan was first to add to the bearing alloy a mere trace of a third metal, which might be supposed to change merely the grouping of the atoms of the other two metals without developing any new crystalline arrangement. Next a somewhat larger quantity of the third constituent was added, which might either develop a new network arrangement of the two excess metals, or develop small crystals of a chemical compound with one of these metals. Then a still larger amount of the third constituent was added, which was thought would bring out a larger number of crystals of chemically combined metals, or in some other way change the microstructure. It was hoped in this way to obtain examples of changes effected by the presence of third constituents in bearing alloys which might in some measure correspond to changes in the higher orders of asperity upon polished sections.

25. Specimens of these alloys were cast in an iron mould in the shape of a bar $\frac{1}{2}$ inch x $\frac{5}{16}$ inch x 4 inches, which was planed to a triangular cross-section, and two sides of the prism thus formed were polished and examined under the microscope, before etching, in order to make out so far as possible the harder portions, which then showed in relief. Then the face was etched and further examined.

26. For the friction test, from a bar like the one whose dimensions were given above, there were shaped two thin sliced bearings, of about the dimensions of the accompanying sketch, which were bolted to the sides of the frame of the testing apparatus above

described, where the bearing surface, *B*, supported the journals at the ends of the spinning shaft, and was the test surface. The journal was run in these bearings until it could be seen just which were the contact areas, and these were then taken down slightly

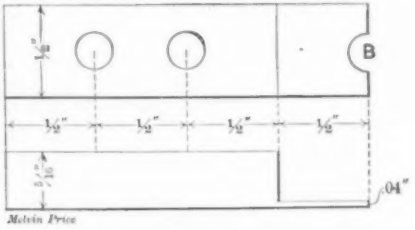


FIG. 241.

with emery cloth and the journals again run upon the bearings, and the process repeated until, so far as could be observed, a perfect fit was attained. Then the retardation curves were taken in the manner indicated above.

27. At first it was believed that the true frictional characteristics of the solid materials would be best exhibited in the absence of unguents. But after numerous trials under such conditions, and also under conditions of scanty lubrication, it was decided that at least for the pressures and speeds commonly employed, the amount of abrasion always present made it seem clear that whatever results might be thus obtained would be some kind of an index of rupture, or inner friction, instead of resistance to sliding or outer friction. Now the author was well aware of the fact that it had often been said that the identity of the solid material was lost whenever flooding of the bearing was resorted to. Thurston seemed to believe this. Charpy had found it so, and noticing the destructive abrasion in the attempts where no unguent was used, thereupon abandoned the friction test. But Professor Goodman's atomic theory was based upon tests where little or no oil was used. The author does not find a complete description of Goodman's experiments, but his own efforts at unlubricated friction determinations have been so absolutely worthless as to make it impossible to place entire confidence in the law so obtained, notwithstanding Goodman's positive statements. Accordingly, in the experiments here recorded the bearings were supplied with as much oil (mineral) as would adhere to the journals and to the test bearings after each one or two performances. Such efforts at giv-

ing the rubbing members as great an amount of the constant unguent as they would "hold" constitute the whole of the efforts made at uniformity of conditions regarding lubrication.

28. Difficulty at once appeared when it was desired to interpret the large mass of data obtained. Uniformity of results at first appeared among the impossibles. The friction of the bearings was variable; worse, it was capricious in its behavior. However, as performances were repeated, great care being taken, with cleanliness and uniformity of lubrication, it was noticed that for a given metal the latterly obtained curves *tended toward uniformity of shape*, although absolute values of resistance always differed, sometimes only slightly, sometimes considerably. In view of this it was decided that certain performances should be elected from the masses, for each metal which should reflect most nearly the frictional qualities of that metal. In electing these performances at least two qualifications were recognized as essential: (1) Recognizing the fact that probably abrasion caused the majority of the caprices above noticed, and that probably always abrasion increased the resistance; and feeling also a desire to find some basis of comparison of metals as to their absolute resistance, it was decided to elect performances from among the *best* for each metal, or those showing the lowest resistances. (2) Whenever there appeared a decided but uniform tendency toward some peculiarity in the variation of the resistance, it was decided to select those performances which emphasized these peculiarities most.

29. As an example, in the case of pure copper, a complete log of whose performances appears above in the discussion of mechanical fit, in the latter part of the log there is noticed a decided tendency toward a lengthening of periods 2 and 3. The running times also show these performances to indicate lower resistances than are indicated by the first portion of the log. Hence it was decided to select as typical of pure copper performances from among those latterly tabulated. Accordingly the last two and the last but three were chosen. Of these, one shows decidedly lower resistance (greater time) during period 3; one shows practically the same resistance in the two periods 2 and 3, and the third shows slightly lower resistance in 2 than in 3. But in the last fourteen performances, eight have lower resistances in period 3 than in period 2, four show the reverse condition, and in two they are tied. Accordingly the last performance but seven, showing marked

anti-friction in 3, was chosen, it being one of the best performances of copper. The last but thirteen was the fifth choice made, and in this periods 2 and 3 are tied. For the ten periods, counting from the point of rest, the chosen performances are set down in the following table:

	Time in dropping from a speed, in r. p. s. of										
	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0	Total.
A.	10	10	11	11	13	15	16	17.5	17.5	12	133
B.	13.5	12.5	11	14	14	16	21	30	17	15	164
C.	11.5	14.5	15	17	20	21	23	29	19	16	168
D.	13	14.5	16	17.5	21	25	27	28	28.5	18.5	209
E.	11	14.5	16	16	18	21	23.5	27.5	31	21	199.5
Number of period	10	9	8	7	6	5	4	3	2	1	178.3*

* Mean of totals.

30. It was now assumed that these were similar curves, *i.e.*, except through error of observation or peculiarity of condition, they differed only in the scale of times. Accordingly they were reduced to the same scale in the following manner: The total running times for each of the five performances were found by adding, in each case, the times for the ten periods. The results are found in the table of totals. The mean of these was taken as a typical total and was made the basis of the reduction scale by which the running times for each period were modified. Thus the mean of totals = 178.3. Accordingly the performance was modified by the factor $\frac{178.3}{133}$ for A, $\frac{178.3}{164}$ for B, and so on, making each horizontal line, when modified, total 178.3. The resulting table of modified performances is as follows:

Performance.	Time in dropping from a speed in r. p. s. of									
	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	13.4	13.4	14.7	14.7	17.4	20.2	21.4	23.4	23.4	16.1
B.	14.7	13.6	12	15.2	15.2	17.4	22.8	32.8	18.5	16.3
C.	11	13.9	14.4	16.3	19.2	20.2	22	27.7	18.2	15.3
D.	11.1	12.3	12.8	14.9	17.9	21.2	23	23.8	24.3	15.7
E.	9.9	13	14.3	14.3	16.1	18.8	21	24.7	27.7	18.8
Totals	60.1	66.2	68.2	75.4	85.8	97.8	110.2	132.2	112.1	82.2
X.	12.0	12.6	13.8	15.1	17.1	19.6	22.1	26.4	22.4	16.4
Number of period	10	9	8	7	6	5	4	3	2	1

31. It was now assumed that the performances, so modified by the scale factors, were directly comparable. Accordingly means were taken of running times for each period (vertical columns), and in the horizontal line, *X*, we have the result. This may be regarded as a mean or composite performance. It is upon this curve, so found, that I have depended for graphical indication for the behavior of frictional resistance of pure copper. Let us see how it compares with the observations made above, when the five performances were chosen from the mass of data in the log. The considerations upon which these were selected are seen to be reflected in the composite curve: (1) The lowest resistance is noticed in periods 2 and 3, the very lowest in 3. The value in period 3, 26.4 in time of running, stands well with the unmodified values. As a numerical check, the total running time in curve *X* foots up 177.9, slightly different from the mean of the totals 178.3, used as a scale basis. The difference may be attributed to neglect of decimals.

32. This method was applied to all metals tested and the following curves plotted:

PURE ALUMINIUM.

Performance.	Time in dropping from a speed in r. p. s. of								
	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	34	39	43	49	52	46	39	25	15
B.	32	41	47	52	59	59	34	30	16
C.	35	40	50	55	65	61	34	30	15
D.	34	40	49	54	56	57	35	35	16
E.	32	37	45	48	60	47	30	32	17
X.	33.5	39.4	46.8	51.6	58.4	53.8	34.5	30.4	15.9
Number of period	9	8	7	6	5	4	3	2	1

PURE LEAD.

Performance.	Time in dropping from a speed in r. p. s. of								
	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	26	19	11	10	13	33	52	76	66
B.	13	11	13	13	13	16	31	59	43
C.	26	25	22	29	38	45	57	98	53
D.	27	12	36	37	40	33	51	77	31
X.	22.9	16.8	20.1	21.5	24.8	31.1	48	79.1	50.9
Number of period	9	8	7	6	5	4	3	2	1

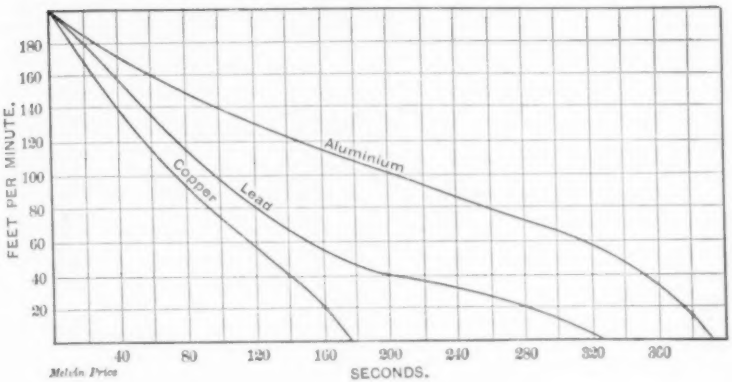


FIG. 242.

ALUMINIUM SERIES.

AL (80). CU (15). MN (5). No. 1.

Performance.	Time in dropping from a speed in r. p. s. of											
	60 to 55	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	18	16	18	20	20	26	28	38	50	50	36	14
B.	....	....	....	23	28	34	38	49	56	57	39	15
C.	....	....	....	20	24	27	34	38	43	33	36	13
D.	....	....	16	19	20	23	26	39	52	51	34	13
E.	....	....	....	....	25	28	34	40	43	38	27	13
X.	....	....	17.3	20.1	23.4	29.5	32.1	40.6	48.6	45.4	34.3	13.6
Number of period.	12	11	10	9	8	7	6	5	4	3	2	1

AL (94.9). CU (5). MG (.1). No. 2.

Performance.	Time in dropping from a speed in r. p. s. of									
	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	22	13	19	12	18	22	24	26	27	16
B.	....	....	19	17	13	20	21	24	23	13
C.	....	....	22	20	17	20	22	26	21	14
D.	18	16	16	15	15	14	19	20	18	11
E.	21	19	19	17	14	20	20	24	21	21
X.	20.6	16.4	19	16.3	15.9	19.1	21.3	24.4	22	14.9
Number of period	10	9	8	7	6	5	4	3	2	1

Al (90). Cu (5). Sb (5). No. 3.

Performance.	Time in dropping from a speed, in r. p. s. of									
	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	15	17.5	15.5	16.5	17.5	17	13.5	15.5	14	11
B.	16	17	19	20	19	19.5	20.5	16.5	15.5	12
C.	15	15	14	13	13.5	14	16	19.5	16	12
D.	13.5	13.5	14	11	11.5	11.5	13.5	15.5	16	12
E.	12	12	13	12	12	11.5	16	13.5	14	12
X.	14.3	14.9	15.1	14.5	14.6	14.5	15.9	16.1	15.3	11.9
Number of period	10	9	8	7	6	5	4	3	2	1

Al (84). Cu (15). Mn (1). No. 4.

Performance.	Time in dropping from a speed, in r. p. s. of									
	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	12	11	12	13	13	12.5	13	14.5	13	12
B.	....	....	....	12	11	12.5	11.5	12	13	12
C.	....	....	....	....	12	12.5	14.5	13	14	11
D.	....	....	....	13	13	14	14.5	16	15.5	11
E.	....	....	12	14	14	15	14	14.5	14.5	13
X.	12.2	11.2	11.7	12.9	12.6	13.3	13.5	14	14.1	11.9
Number of period	10	9	8	7	6	5	4	3	2	1

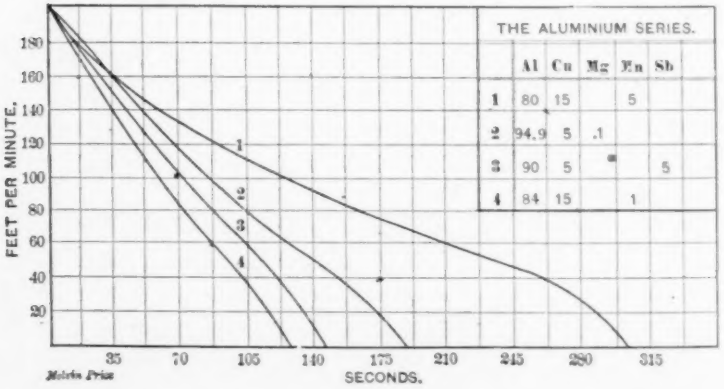


FIG. 243.

THE TIN SERIES.

Bi (.1). Sb (10). Sn (89.9). No. 1.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	17	20	24	22.5	21.5	28	29	35	20	21	20
B.	20	17	17	20	24	25	33	31	37	43	35
C.	....	17	20	22	26	27	32	34	46	53	39
D.	....	17	20	20	25	25	25	33	25	29	46
E.	....	21	22	22	26	30	29	33	33	58	43
X.	20.1	18.6	20.9	21.6	24.6	27.3	29.8	33.7	31.7	39.8	36.4
Number of period	11	10	9	8	7	6	5	4	3	2	1

Cu (4). Sb (12). Sn (84). No. 2.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	14	15.5	18	17	19	21.5	23	25.5	21
B.	14	15	20	19	23	25	28	32	31	37	22
C.	....	....	22	26	28	33	37	39	39	41	25
D.	....	....	19	22	26	28	32	32	32	30	20
E.	....	11	17	17	21	21	25	30	37	38	15
X.	....	13.8	18.5	19.9	23.3	24.7	28.1	30.9	32.2	34.6	21
Number of period	11	10	9	8	7	6	5	4	3	2	1

Bi (5). Cu (4). Sn (91). No. 3.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	15	15	16	17	17	19	21	21	25
B.	....	10	10	12	13	16	16	20	23	26	27
C.	....	....	....	10	7	8	18	13	24	28	29
D.	....	....	17	18	20	24.5	25.5	29	31	36	23
E.	....	....	....	20	20	24	25.5	29.5	36	37	25
X.	....	10.9	14.3	14.6	14.8	17.4	20.3	21.6	26.9	29.5	26.7
Number of period	11	10	9	8	7	6	5	4	3	2	1

Bi (5). Sb (10). Sn (85). No. 4.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	....	17	15	17	16	17	30	32	20
B.	9	7	11	11	9	10	13	14	17	20	24
C.	....	13	9	7	7	8	5	14	23	23	26.5
D.	10	13	18	19	17	19	21	24	25	30	26
E.	....	8.5	9.5	12	12	13	17	19.5	23.5	31	28
X.	9.6	11	12.2	12.9	11.7	13.1	13.9	17.7	24.1	27.3	25.9
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Bi (.1). Cu (4). Sb (95.9). No. 5.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	10	8	9	10	10	10	14	19	27	20
B.	....	....	13	12	12.5	13.5	13	15	17	17	16
C.	....	10	11	12	13	13	15	16	20	21	15
D.	....	....	11	14	13	15	16	19	20	23	18
E.	....	10	12.5	10.5	8.5	7.5	7.5	16	20.5	22.5	19
X.	....	10.5	11.3	11.6	11.3	11.6	12.2	16.2	19.6	22.5	17.9
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Bi (1). Cu (4). Sn (95). No. 6.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	13	12	12.5	13.5	13	15	17	17	16
B.	....	....	11	14	13	15	16	19	20	23	18
C.	....	10	12.5	10.5	8.5	7	7.5	16	20.5	22.5	19
D.	....	13	12	12	10	13	17	18	19	22	20.5
X.	....	11.7	11.9	12.1	11	12	13.2	17	19.2	21.2	18.4
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Bi (1). Sb (10). Sn (89). No. 7.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	....	7	8	9	9	8	11	13	17
B.	....	....	12	9	8	8	8	7.5	7.5	14	17
C.	....	....	....	15	14	12	13	16	10	13	19
D.	....	11	8	9	10	10	12	20	14	15	27
E.	....	....	....	....	....	16	16	17	20	21	25
X.	....	9.5	11.7	11.1	11.0	11.0	11.5	13.3	14.4	15.5	21.2
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Sb (10). Sn (89.9). Zn (.1). No. 8.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	....	8	8	10	8	10	12	17	13
B.	....	....	....	8	8	10	8	10	12	17	13
C.	....	....	5	7	6	7	12	15	18	21	16
D.	7	6	6	8	6	8	8	9	9	16	12
E.	7	7	12	6	8	7	9	10	15	19	15
X.	....	7	7.7	7.5	7.3	8.5	8.9	10.7	13.0	18.1	13.8
Number of period.....	11	10	9	8	7	6	5	4	3	2	1

Sb (10). Sn (89). Zn (1). No. 9.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	....	14	13	12	13	13	15	17	15
B.	....	....	14	14	12	14	14	14	16	18	16
C.	....	....	15	15	15	15	16	16	18	19	16
D.	....	15	14	15	16.5	15.5	15	17	19	21	17
E.	....	....	18.5	15.5	16	15	16	17	20	20	18
X.	....	14	14.9	14.5	14.5	14.3	14.8	15.3	17.6	19.0	17
Number of period.....	11	10	9	8	7	6	5	4	3	2	1

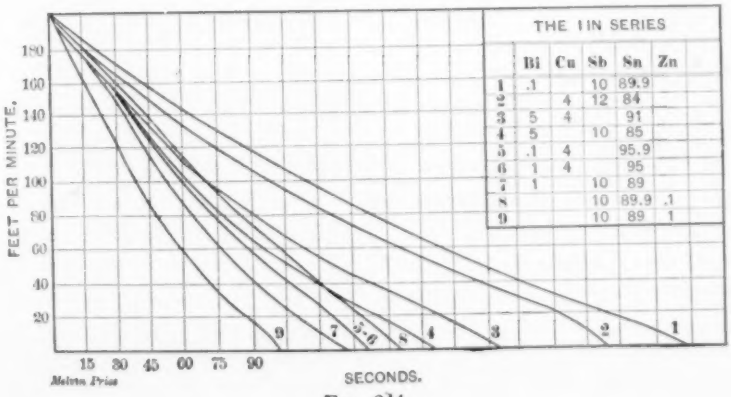


FIG. 244.

MISCELLANEOUS ALLOYS.

Pb (80). Sb (15). Sn (5). No. 1.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	20	26	23	28	25	32	35	43	27
B.	....	20	20	19	24	25	27	30	30	38	30
C.	....	....	17	15.5	19.5	21	24	26	31.5	33	20
D.	....	....	15	13	12	13	13.5	24.5	53	51	26
E.	....	12	11	12	14	23	22	37	49	54	25
X.	....	14.7	16.7	17.0	18.9	21.8	22.2	30.4	40.1	43.8	25.5
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Al (10). Cu (89.9). Mn (.1). No. 3.

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	8	8	9	9	9	9	10	10	10	9
B.	....	....	9	10	9	10	10	10	9	9	8
C.	....	....	....	9	9	10	10	10	10	10	9
D.	....	10	10	10	11	11	11	12	11	11	11
E.	....	....	....	....	9	9	10	10	11	10.5	10
X.	....	8.6	8.9	9.4	9.4	9.8	10.0	10.4	10.2	10.3	9.4
Number of period.	11	10	9	8	7	6	5	4	3	2	1

Al (10). Cu (89). Mn (1). No. 2.

Performance.	Time in dropping from a speed in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	10	8	10	10	10	10	11	11	10
B.	....	....	....	11	11	10	10.5	12.5	12.5	12.5	10
C.	....	....	10	11	10	10	10	10	10	10	8
D.	....	9	9	10	9	9.5	9.5	9.5	10.5	10	10
E.	....	....	....	....	8	9	9	9	10	9.5	9.5
X.	....	9.3	9.8	9.9	9.6	9.7	9.8	10.2	10.8	10.5	9.5
Number of period.	....	10	9	8	7	6	5	4	3	2	1

A (5L). Cu (75). Mn (20).

Performance.	Time in dropping from a speed, in r. p. s. of										
	55 to 50	50 to 45	45 to 40	40 to 35	35 to 30	30 to 25	25 to 20	20 to 15	15 to 10	10 to 5	5 to 0
A.	....	....	....	10	10	9	10	9	11	10	9
B.	....	....	....	9	9	9	9	10	10	11	10
C.	....	....	9	10	9	9	10	10	11	11	9
D.	....	....	9	11	10	10	10	10	11	10.5	9.5
E.	....	....	....	....	10	10	10	10	10	10	8
X.	....	....	9.1	10.1	9.7	9.5	9.9	9.9	10.7	10.6	9.2
Number of period.	11	10	9	8	7	6	5	4	3	2	1

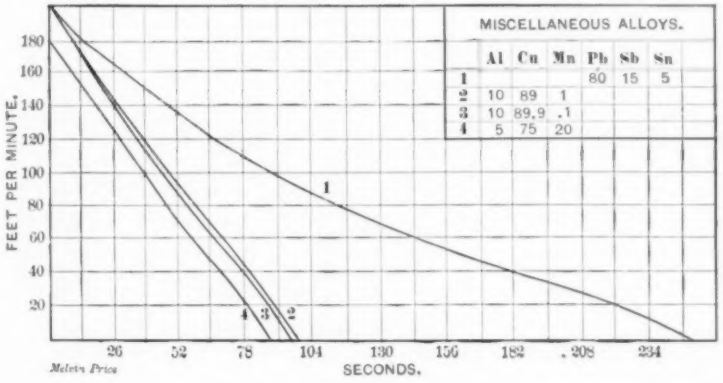


FIG. 245.

33. *Curves showing the Variation of Frictional Resistance with the Speed.—Noticing first the curves of the pure metals*, it is at once apparent that aluminum, copper and lead have similar speed friction variations in general, but their curves are quite different in detail. Generally, in all three, the resistance is high at high speeds, becoming less as the speed lowers, but rising again near the vanishing point of speed. In detail, at high speeds, there seems to be no striking differences, except in the amount of resistance, but at middle speeds lead exhibits faster variation than aluminum and copper, although it is between them in average amount of resistance. Between aluminum and copper the difference noted is that the variation of aluminum is much more gradual, or its resistance is most nearly constant of the three. It is at the lower ends of the curves that the most striking differences in detail are noticed. The aluminum comes down abruptly when 20 ft. per

min. is reached, lead consumes more than three times as much time in coming to rest, while copper, though not so abrupt with reference to the other curve, is not very different from aluminum in actual time of coming to rest from the above mentioned speed.

34. The resulting curves for these three metals are not at all similar. A scale factor applied to any one which would make the total running times equal would show the other coördinates vastly unequal. This has the effect of impressing us with the fact that the metals have frictional individualities.

35. In the *aluminum series* we are first impressed by a general similarity at low speeds, though the abruptness differs greatly in degree; a similarity at middle speeds, but in this instance it is constancy and not variation; and by some peculiarities exhibited at the higher speeds. Curves 1 and 2 have opposite qualities at high speeds, while 3 and 4 are similar, constant like their middles, and unlike 1 and 2. May this difference be attributed to the fact that in 1 the third constituent is manganese and in 2 it is magnesium? And if so, is the fact due to abrasive conditions, or effects, or to some one of the orders of asperity? More will be said about this matter in the later discussion of results.

36. In the *tin series* we have a greater variety of third constituents. At the high speed the curves differ little except in direction, being nearly straight lines. Nos. 2, 8 and 9 have noticeably reverse curvature at lower middle speeds, with the abrupt down-turning at very low speeds. Of these 2, which is Babbitt's metal, has the gradual variation of the aluminum series. Nos. 8 and 9 are entirely similar, differing only in the value of the resistance, indicated by their slopes, and it is seen that their compositions differ only in the proportions of the third constituent, zinc.

37. For the first time we now come across curves which exhibit no sign of abrupt change of resistance in passing from lower middle to low speeds. Nos. 1, 3, 4, 5, 6 have nearly the same slopes throughout these low speeds. In 7 there is even a slight opposite or upturning tendency in contrast with the curves we have heretofore examined. In the table it may be noticed that these alloys all contain bismuth, but in varying proportions and alloyed with copper and tin in three instances and with antimony and tin in the others.

38. Consideration of the tin series tends to confirm the belief in the general principle that composition determines frictional char-

acteristics, since the peculiar shape of curves 8 and 9 have occurred only when zinc was a constituent and that of 1, 3, 4, 5, 6 and 7 only when bismuth was present. Admitting that in all probability there is no method of measuring absolutely the frictional resistance, it may be interesting to note, nevertheless, that the only difference in the similar curves is one of slope, and that the only difference in the composition is the proportion of the third constituent. Here the third constituent may be regarded as either of the two low percentage constituents, so that without some discussion of the microstructures we may not even guess which of the two influences the slope. In this connection we may notice, however, that curves 5 and 6 coincide throughout: and that their compositions differed only in the proportion of bismuth, the percentage of copper being equal; while 3 and 4 have the same percentage of bismuth, different slopes throughout, and different third constituents, viz., Cu and Sb. Similarly for curve 6 and 7. These characteristics contribute toward the impression that, of the two low percentage constituents, one contributes the shape of the curve or determines its law of variation with speed, while the other determines the absolute value of the resistance at any given speed. To use a mathematical expression, *one constituent determines the law of the variation, and the other the constants of that law.*

39. Looking at the sheet of curves taken from miscellaneous alloys, we have in 1 a long, sweeping curve, nearly straight at high speeds; more variation, but gradual, at medium speeds, and considerable reverse curvature at lower speeds, with a down-turning at the lowest points. This alloy is rich in Pb. Reverting to the curve of pure Pb we see striking similarities. Now antimony and tin are the other constituents of 1. And it is thought that these form the chemical compound of SbSn. In the absence of other data we might say that probably the difference in resistance noted is a consequence of the presence in the alloy of grains of this chemical compound, which is vastly different in some physical properties from the Pb of the matrix.

40. In the other three curves on the sheet marked "Miscellaneous Alloys" we have a portrait of the frictional behavior of the yellow alloys of aluminum and copper. Here we may regard the manganese as the third constituent. The curves, 2, 3 and 4, nearly coincide. The greater portions of 2 and 4 are straight, while 3 has a slight tendency toward downward convexity. All have the reversal at lower middle speeds and a slight tendency to

abrupt down-turning at low speeds. Curve 4 is that of an alloy having 20 per cent. of manganese, while in 2 and 3 the percentage of manganese was respectively 0.1 and 1. This makes it appear that in these instances the percentage of the third constituent has had little or no effect.

41. *Microstructures.*—In the writer's laboratory at the University of Nebraska, about one year after the above resistance data were taken, the alloys were examined with the microscope. The original ingots, which had been shaped into bearings for the resistance tests, were sectioned, polished with rouge and water upon a wheel faced with baize, etched by a light acid attack, and magnified for examination.

42. If we are to look for confirmations in the microstructures of any beliefs suggested by frictional performances, we must, of course, neglect the effect of the year's time which elapsed between the tests and the examinations which will now be recorded. It has been established that changes in granulation occur after solidification. It was notably established also by Roberts-Austen, in his gold-lead experiment, that solid solutions may be formed without resorting to liquifaction by heat. However, the writer made examinations of the above alloys concomitantly with the frictional tests at the Columbia University laboratory, and so far as he is able to remember, nothing found then had vanished when the work was taken up a year later, although many pecularities were revealed in the latter examinations which did not reveal themselves the year before. But these, it is believed, are due to the more leisurely manner in which the later work in microscopy was performed.

43. Considerable uncertainty exists regarding the chemical combinations of metals. It is widely believed that SbSn exists. Also Cu_3Sn and $SbCu_2$. Indications in the work of Alder Wright have pointed to Bi_3, Sb_2, $CuZn_2$, and $CuZn_4$. In the absence of more reliable data, it will be assumed that some such compounds exist, but the atomic proportions will be omitted. Thus all compounds believed to exist will be designated by writing the symbols without subscriptions, as SbSn, CuSn, SbCu, etc. By so doing the writer does not mean to subscribe to the belief that the chemical compound CuSn exists or that AlCu exists, as some profess to believe.

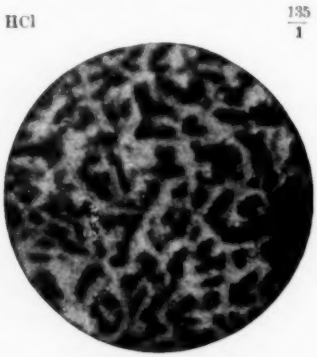
44. It must also be remembered that we are here dealing with ternary alloys. Very great uncertainty exists as to just what happens in a given case when three metals are melted together and,

after being mechanically mixed, allowed to freeze. The work of Alder Wright, as published in Prof. Roberts-Austen's "Introduction to the Study of Metallurgy," is not complete enough to throw any light upon the cases here in hand. But generally, in the present investigation, one of the constituents has been given such a small proportion of the whole mass, that the alloy may presumably be regarded as a binary alloy.

45. In ideas of friction, hardness of the rubbing members comes largely into evidence. And in alloys, where it is expected that totally different physical substances will occur side by side, it is the *differences* of hardness that we look for largely. Thus it interests the investigator of friction to know whether or not there are in a given alloy hard grains embedded in a soft matrix, or soft islands surrounded by a hard network, or "gridirons" made of interstratified plates, soft or hard. The author has by two methods only been able to draw inferences about differences of hardness: (1) By the use of the sclerometer. (2) By observation under the microscope during the operations of polishing the surfaces before etching. The first kind of observation was found useful in the case of the "embedded hard grain" alloys, for it was found easy to detect deviations in the path of the diamond point where it had passed around, or run over without deeply scratching, the hard grains. The second kind of observation was resorted to in a few such cases and generally confirmed the sclerometer tests. In the case of the "hard network" alloys, relief polishing, by the second method, was the best resort. Whenever an alloy, after being polished carefully in relief, was found to have dark areas when examined microscopically, it was assumed that out of these dark areas there had been brushed a softer or else a weaker material.

46. When no means was at hand of knowing what differences of hardness existed, it was assumed that portions of the surface which behaved differently when treated with acid were of different constitution and therefore probably differed in hardness. As an example, Fig. 246 shows the photomicograph of an aluminum-copper alloy, in which magnesium is the third constituent. Relief polishing gave no evidence of differences of hardness, and neither did the sclerometer. But when etched, large black islands formed on the surface. The planimeter shows that the islands constitute almost exactly one-half of the area. Neglecting the magnesium, it is evident that if this black portion is the chemical compound

of aluminum and copper, it must be one in which the number of atoms of aluminum is very great, compared with the number of copper atoms; in fact, the disparity is too great to admit of the possibility of its being composed entirely of such a compound. The same may be said of the white area. If the alloy is a solid solution of copper in aluminum, it is a very heterogeneous one, and marked segregation has taken place where we should expect the opposite under the conditions of casting (the ingot was cast in a chill mould). It is at once seen that if either of the eutectics Al (67) Cu (33) or Al (46) Cu (54) had formed the area taken by it would not be nearly so large as one-half of the total area.



Al (94) Cu (5) Mg (1)

FIG. 246.

47. Thus we are at sea regarding the constitution and hardness, though we might say, "if the constituents differ, the hardnesses differ." But when we come to examine this alloy in connection with others of similar constitution, we are somewhat enlightened.

48. *The Aluminium Series.*—In Fig. 247 are shown the white alloys of copper and aluminium, resistance curves of which have been given above. The compositions of these differ little from that of Fig. 246, lately discussed, the latter being nearly the same as (b) of Fig. 247. But these were etched with HNO_3 instead of HCl. To illustrate more clearly the difference of appearances when differently etched, Fig. 248 shows (b) of Fig. 247 etched with HCl. The thin medial lines in the white network of Fig. 248 are not unlike the black lines of (b) Fig. 247, and we may reasonably conclude that they are the same. Something seems to have caused the acid to begin the attack on the aluminum crystals at their

mid-points and has preserved from attack the outside portions near the black lines.

49. Despite this study in etching, the decision regarding hardness is still attended with uncertainty. Before etching the polished surfaces of the four alloys were examined microscopically.

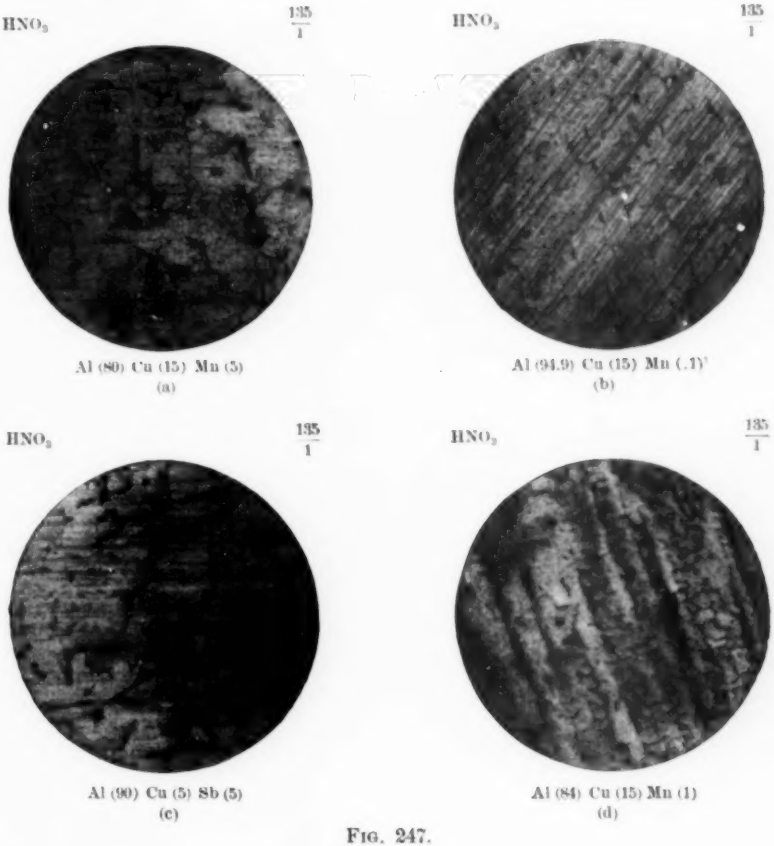


FIG. 247.

At first no structure appeared visible on the surfaces of (a) or (b), and the larger straighter markings of (c) appeared as black blotches, but when (d) was put under the glass, a faint white network appeared, which corresponded closely to the black lines in the etched view of (d), and which appeared to be in relief. When the others were re-examined for this white network feature (a) showed very faint signs of such, but (b) did not, and the straight

black blotches only characterized (c). It is believed from this that the black crooked lines, which show in all four alloys, are of the harder substance (possibly AlCu), and that the straighter black blotches of (c) are of a softer or weaker substance.

50. Excepting that we may attribute the last named feature to the antimony in the case of (c) there is nothing but conjecture about the effect of the third constituents of (a), (b) and (d). They do not seem to assert themselves excepting in (a) and (b), the resistance performances of which differ so markedly at high speeds.

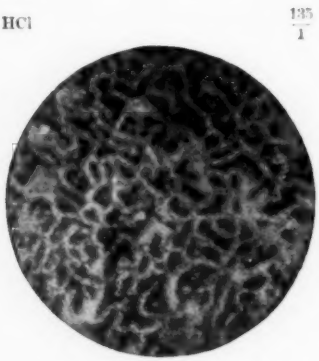
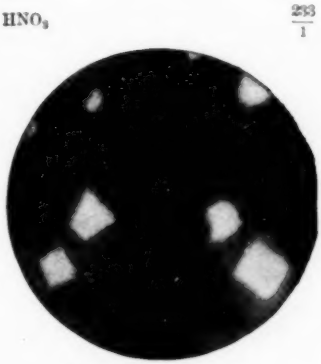


FIG. 248.

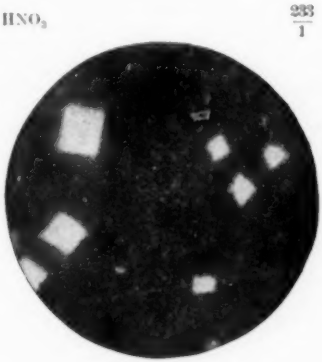
The Tin Series.—In Fig. 249 are photomicrographs of the six alloys of the Tin Series which contained bismuth. All of these are shown by the sclerometer to have hard white crystals, embedded in soft matrices. These crystals are of three types. (1) Nos. a, d and g have square white patches, and since these alloys have goodly proportions of antimony it may be concluded that these are cubes of the compound SbSn. They seem much harder than the matrix which latter we must believe to be largely the pure tin. (2) Nos. (c) and (e) are seen to have long, white, slender patches, as well as small dots and chains of dots. (3) No. f has the same kind of small dots, but no chains or needles. The white crystals in c, e and f are harder than the matrix, so far as the writer is able to discover. These are probably CuSn.

51. Referring to the curves on the sheet marked "Tin Series," it has already been remarked that all alloys containing bismuth had the featureless endings at the X-axis. These curves are practically similar, and hence there is nothing in them to reflect the

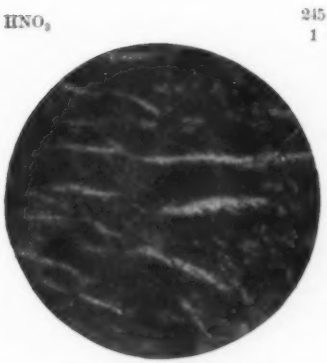


Bi (.1) Sb (10) Sn (89.9)
(a)

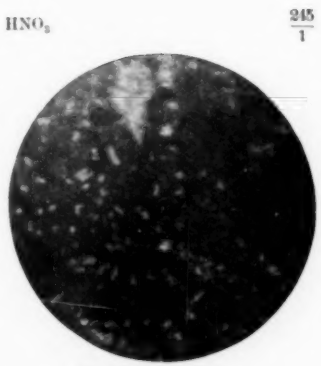
Bi (5) Cu (4) Sn (91)
(c)



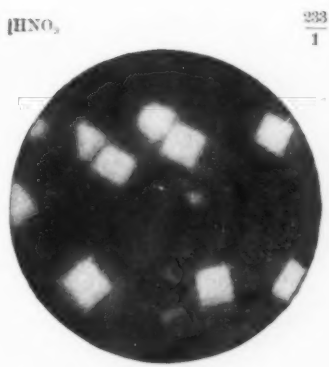
Bi (5) Sb (10) Sn (85)
(d)



Bi (.1) Cu (4) Sn (95.9)
(e)



Bi (1) Cu (4) Sn (95)
(f)



Bi (1) Sb (10) Sn (89)
(g)

FIG. 249.

difference in shape and size of the crystals above remarked in the microstructures.

52. Below, in Fig. 250, are shown photomicrographs of the remaining three alloys of the tin series. All contain white squares,

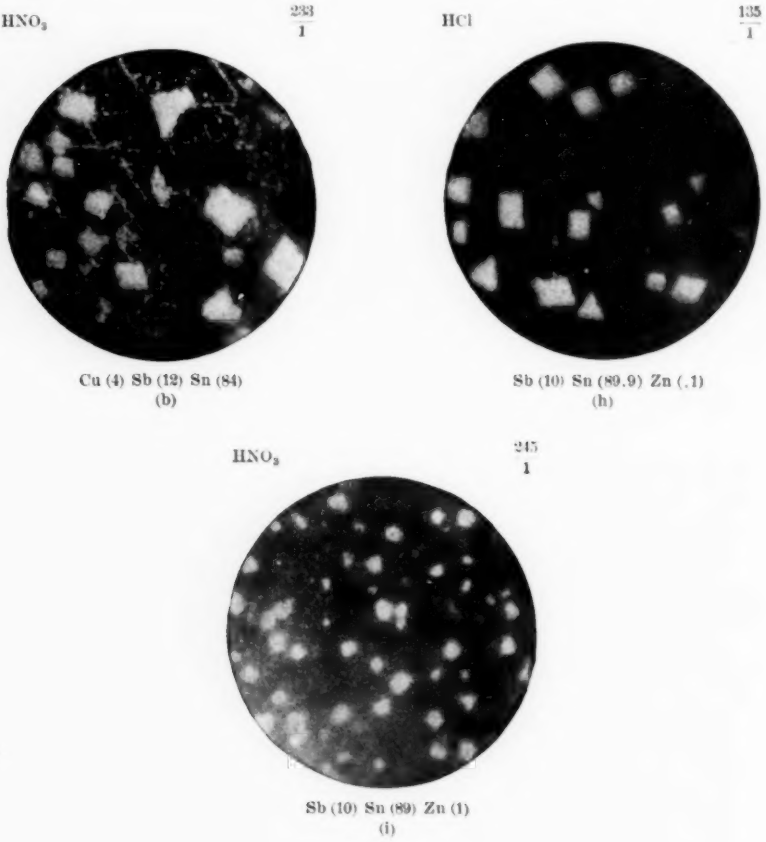


FIG. 250.

which we interpret to mean hard cubes of SbSn. No. (b) is Babbitt's metal. It has white needles (presumably CuSn) in addition to the cubes of SbSn. Nos. h and i, which contain zinc as the third constituent, seem to have no hard granules except those we have called SbSn. It is seen that with Nos. b, h and i we might attribute the similarity of curves h and i, and their dissimilarity to (b), to the difference in microstructures, viz., the absence of

needles and dots among the white cubes in h and i. But notice that i is magnified nearly doubly as high as h, and that even then its cubes are not nearly so large. Actually, then, the cubes of i are but dots compared with those of h. The *size* of the granules seems to have no reflection in the *shape* of the curve. Hence we can scarcely attribute the dissimilarity of b to h and i to the small dots of the former.

53. The great question is, to what *shall* we attribute similarities of curves, so that our conclusions may be confirmed by microstructure. Looking at the nine curves, and the nine structures of the tin series, as a whole, all have hard grains embedded in soft matrices. But the curves present many differences in shape. (1) So that hard grains so found are not themselves sole causes of any peculiar resistance variation. (2) Also, the size of the grains seems not to determine this resistance variation. (3) Nor does the shape of the grains themselves seem to have the determining effect on the shape of the resistance curve. But let us now notice those constituents which we have not mentioned in discussing the structures, because there seemed no good reason for their appearance as crystals, such as have been mentioned as the features of the microstructures. These may be called the "silent constituents," and are exemplified in the bismuth of a, c, d, e, f and g, and the zinc of h and i, which do not "show up" in the structures. Upon failure to assign reasons, obtainable from structure, for peculiarities of curves, may we attribute the latter to the unseen effect of the "silent constituents"? The remarkable similarity of all curves of alloys containing bismuth and of those containing zinc lend considerable weight to this otherwise purely circumstantial evidence. We may say at this time that we have here part proof, and as yet no reputation, of some such law as the following: *The effect of constituents which form hard crystals is constant; the effect of "silent constituents" varies with the speed.*

54. Looking at the photomicrograph of the alloys whose curves have been given on the sheet entitled "Miscellaneous Alloys," we have the structural features of "white metal" and of three yellow alloys of aluminum and copper. See Fig. 251. In the white metal we detect small squares, and seeing that the composition admits that possibility we may say these are cubes of SbSn. These crystals are comparatively few in number, so that the structure would indicate a resistance performance not very different from that of pure lead. This is seen to be the case when curve 1 of "Miscellaneous"

sheet is compared with the curve of lead on the sheet of pure metals. Here is more verification of the belief that crystals of chemically combined metals embedded in a softer matrix do not affect the manner in which the resistance varies with the speed. A question of great industrial importance is whether or not these

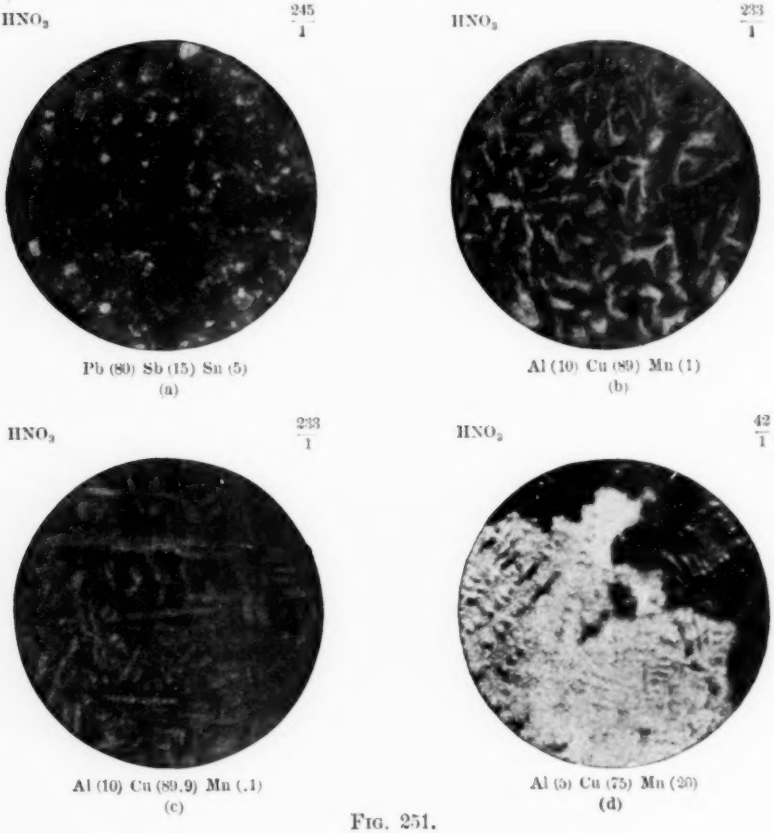


FIG. 251.

crystals greatly affect the actual value of the resistance in the constant manner intimated above in the discussion of the tin series. Comparing the curves of "white metal" and pure Pb, by the scale of times, such belief is confirmed, for the total running time of "white metal" was 252 seconds, while the performance of pure lead was 332 seconds in duration. The effect of the crystals seems to be just what might be expected, *i.e.*, that of increasing the resistance as a whole.

55. The microstructures of the "aluminum bronzes" are of a totally different kind from any heretofore presented in this account. When etched these specimens all showed remarkably well defined crystals, closely packed together, without any well ordered arrangement, and differently tinted by the acid, making the surface very mottled in appearance. A very noticeable feature was the gridiron-like appearance of many of these crystals themselves. Each one seemed to be a stack of plates set edge on to the surface, but with direction peculiar only to that crystal. No. d shows this best, though the resemblance can be traced in c and to some extent in b. None of these shown exhibit the "gridiron" feature quite so well as it appears in the macrostructure, without magnification. Aside from the faint resemblance noticed above by the aid of the microscopic observation, there is little of similarity to be noticed in the microstructures. Yet the frictional performances were almost identical. So that here we have a claimant for the position we have heretofore assigned to the "unseen constituents." Is it then microstructure only or, in terms of friction, the lower orders of asperity which determine the resistance variation? Now these lower orders of asperity could hardly have been the cause of the "white metal" pure lead similarity of performance. Again, the "silent constituent" was the same in all three of these latterly discussed alloys, and therefore still have as much right to the importance which has been attached to them as they had before.

56. *Conclusion.*—We may now apply the knowledge gained from the study of the resistance curves and the microstructures to the three questions named in the first part of this paper. They were as follows: (1) The law of variation of the amount of friction with the velocity of sliding. (2) The theory of Charpy and others regarding the function of the hard grains and of the plastic ground-mass in the microstructure of alloys. (3) The theory of Prof. Goodman, having to do with the atomic volumes of the metals employed in bearings.

57. (1) A glance at the results shows that a "law" could be made to apply to all of these cases at once only in an extremely general way, if at all; and that, when we consider details, nearly every affirmation in such a law would be contradicted by some of the results, leaving almost nothing in the nature of a general law of variation, applicable to all alike. In short, an alloy's resistance performance seems to be peculiar to itself, although there are often partial similarities.

58. Henceforth our thought may take two or more different channels. We may try to explain matters by the aid of the microstructures (which should include "silent constituents"), or we may attempt to account for the variations noticed upon some purely mechanical or, perhaps better expressed, accidental conditions. The microstructures, compositions, etc., having been discussed above, let us see what we may do with abrasion as our light. It is admitted that generally abrasion is greatest at higher speeds, as we should expect. The upper portions of the curves would generally support an abrasion theory of the variation. There is also support for this in the fact that the softer metals show greater changes of resistance in falling down from the higher speeds. But the latter statement is controverted very largely by comparisons in this detail. Take the pure metals, aluminum, lead and copper; aluminum, which is second in hardness, has very little variation, while copper, first in hardness, has nearly as much of this peculiarity as the very soft lead. As a last argument in support of the influence of abrasion, we may cite the curves of the aluminum-bronzes. These were the hardest of the alloys, and show the least—practically nothing—of variation. So that, for these instances, where there was least abrasion, there was least of higher-speed variation of resistance.

59. At the lower ends of the curves there are two distinct peculiarities. Fourteen of the twenty curves are decidedly convex upward, and the remaining six just as decidedly straight or convex downward. Abrasion can hardly explain the difference. Not only can it not explain the upward convexity of the fourteen, but that peculiarity operates against such a theory. And meantime the straightness of the other six lends absolutely no support to an abrasion explanation. Also, against such an explanation there may be cited numerous small differences, noticed at high speed, between alloys physically very much alike. As an example of this we have the marked difference at the high speeds between curves 1 and 2 of the Al series. Moreover, in general, the peculiarities of the curves were most marked when the average resistance was least, *i.e.*, when the total running times were longest, which fact operates effectively against the soundness of the abrasion argument, for abrasion at once increases the resistance.

60. On the other hand, it appeared that mechanical imperfections, microscopic structural features, hard crystals of chemical compounds and other "foreign" effects, which make up the *lower*

orders of asperity, had a rather constant influence on the resistance. Also, that groups of alloys could be formed in which peculiarity of performance went hand in hand with one of the three constituents. This constituent was not noticed in the microstructure and was called the " silent constituent." Accordingly it may be inferred that such constituents affect the grouping of the atoms, or even the minute crystals, in such a way as to determine the feature of the *higher orders of asperity*.

61. On the whole, it does not seem at all improbable to the writer that the lower orders of asperity exert a nearly or quite constant resistance, or that the higher orders exert resistances varying according to the microstructure, although it may also be true that the variations noticed are resultants of two or more effects, either of which might be constant if taken separately, but that under varying conditions of speed at points more and at other points less of each effect enters into the resultant.

62. (2) Charpy's conclusion seems to be that it is the hard grains that contribute to the alloy whatever anti-friction quality it possesses. The results of the present investigation seem to the author to work against that conclusion. Notably so in the case of pure lead and the " white metal," which is largely lead, but has the hard cubes of SbSn. While these results are not intended in any sense to be regarded absolute measurements of resistance, still it may be well to note the performance of the pure metals, aluminum, copper and lead, in comparison with alloys of those metals. Aluminum ran longer than any of its alloys; and so did lead; likewise copper performed with much less resistance than did any of the yellow alloys of aluminum with copper. The only conclusion which can be come to is that the foreign substances introduced into an alloy as a result of the formation of isolated hard crystals tend to increase the resistance, and as noted in (1) above, in a somewhat constant manner.

63. (3) The conclusion of Prof. Goodman was that if the third constituent of an alloy were a metal of high atomic volume, the resistance was diminished, and if of low atomic volume, it was increased. The observation of the writer that these third constituents when " silent " seem to have a great influence upon the variation of resistance with the speed, leaves the way clear for such a conclusion as Goodman's to obtain: since we may assume that the atomic volume of the element effects the higher orders of asperity.

64. In the actual results, look at the curves of the *tin series*.

Comparing 1 and 2, it is seen that in 2 the bismuth of 1 has been replaced by copper. The result is slightly in favor of Goodman's theory, for the atomic volume of Bi is 21.1 and (1) ran 285 seconds, while that of Cu is 7.2 and 2 ran 248 seconds. But the reverse result is obtained when in 6 we replace copper by antimony (atomic volume, 17.9), obtaining 7, which showed greater average resistance. So that these results do not support Goodman's. But it is realized that what is now offered can only be regarded as meager in its power to refute or confirm.

Discussion.

Prof. Melvin Price.—The writer intends that the investigation, so far as it has proceeded, will be regarded as but tentative. He also hopes that all very positive statements, unless they are easily demonstrable by facts, be thought of as opinions prompted by experience, but which may extend too far, or may seek to apply to some case principles not applicable thereto, or in short, be in error. In every such case the writer regrets being unable to revise with language more reserved.

The discussion of laws (1) and (2) in the first portion of the paper is merely the outcome of a desire on the part of the writer to do what he may toward some reconciliation between the work of Morin and that of the more recent investigators. The work of Morin seems too extensive to be ruthlessly thrown aside as it has been by many, especially engineers. For, on the other hand, physicists and mechanicians largely adhere to Morin's results. Such a strife produces in the mind of the average engineering student an unenviable but inevitable opinion regarding present-day knowledge of friction.

The digression into laws (1) and (2) is not in itself highly defensible, nor are the statements advanced thought to be invulnerable. At points the argument is essentially conjectural, although believed by the writer to be approximately true.

He would especially warn those who may think to apply these results practically that none of the measurements taken in the investigation should be regarded as absolute. Similarities and dissimilarities among the curves were especially looked for, but only ordinary precautions were taken to promote that constancy essential to standardization for absolute results. Again, only

the nature of the stationary rubbing member has been taken up, the moving member having been the same for all tests, and as nearly constant in condition as has been ordinarily possible. This journal was probably not widely different from the average practical shaft journal. Nevertheless, much experiment would still be necessary before the results, even if standardized, could be applied to computations.

Mr. Sanford A. Moss.—I do not think Prof. Price correctly states the modern belief regarding frictional behavior. The best statement of what seems to me to be the modern belief is given by Mr. H. F. Moore in a thesis presented to Sibley College, Cornell University, and reprinted in the American Machinist beginning September 10th, 1903. Mr. Moore gives three kinds of friction:

1st, friction of oily solid on oily solid. Mr. Moore calls this "solid friction." Morin's experiments probably cover this kind.

2d, friction in which there is a complete film of oil between two solids. Then the solids are nowhere in contact with each other so that their character has nothing to do with the matter, but all the action depends on the oil. This is called "perfect film friction," and is a matter of hydrodynamics and not of metallurgy. Professor Osborne Reynolds has treated this matter extensively.

3d, to quote Mr. Moore "a stage intermediate between the two above classes, in which there is not a perfect film separating journal and bearing, and yet in which the friction is not so great and does not follow the same laws as in case one." Mr. Moore calls this "intermediate friction."

All of Prof. Price's work falls in the third class mentioned above, "intermediate friction." In intermediate friction the matter is to a great extent dependent on the way oil is supplied, as may be readily imagined. It is probably for this reason that there have been such discordant results obtained by various experimenters, and that Prof. Price does not come to any definite conclusion. Using the figures Prof. Price gives, I find he has had a bearing pressure of about 250 pounds per square inch, and an initial rubbing velocity of about 200 feet per minute. These conditions are rather extreme in bearing practice, and if they existed in an actual bearing would call for very considerable care in lubrication. The bearing pressure is far greater than that which would permit of perfect film lubrication. As Prof. Price states, his lubrication was far from satisfactory, and this probably accounts for many vari-

ations. In the region in which Prof. Price was working the composition of the metals is only one factor, the character of the lubrication being another very important consideration.

It seems to the writer that consistent data for the case of intermediate friction can only be obtained when there is complete flooding of the bearing with oil, so that the bearing surfaces have as much oil as possible between them at all times.

The President.—One of the interesting features of this paper is that the contribution comes from the laboratory of the University of Nebraska. Few of us realize the present activity of scientific research in some of these western colleges. I have happened to be brought in contact with the scope and quality of research for engineering data carried on in the laboratories of two or three of these State universities and have been much impressed by this tendency of the times—we shall soon be looking for newest data in certain lines no more in the output of old and famous laboratories of Germany, France and England, than in the returns from Wisconsin, California, Michigan, Kansas and Nebraska.

Mr. F. W. Taylor.—I am very glad to hear our President make what may be called encouraging remarks on this paper. It seems to me that the investigations made by the author have been thorough and painstaking, and that he has succeeded in getting something out of it which is, to say the least, exceedingly interesting. The paper also contains, as I read it, a promise of good work in the future from the same author. The written criticism is perhaps just. I think probably the author might now be willing to accept a lubrication as suggested by the last writer as better than the condition which he took, but to allow adverse comment to remain the only criticism on thorough and painstaking work of this sort seems to me rather unfortunate. It does not encourage experiments of this sort, and even granting that this particular field may not prove especially fruitful in practical results, similar investigations are of extremely great value to the engineering fraternity. I have not the pleasure of the acquaintance of the gentleman who has presented this paper, but I have been greatly impressed by the thoroughness and care with which his work has been done.

Mr. H. H. Suplee.—In connection with this paper, I should like to call attention to some very thorough investigations on the friction of bearing metals which were made in Germany in connection with the trials of high speed electric railway motors. Mr.

Lasche, one of the engineers of the Allgemeine Elektrizitäts-Gesellschaft, was in charge of the trials, and he made a very exhaustive set of experiments in order to get the satisfactory bearings for high speed cars, running at 125 miles an hour. The result of those experiments have been published very completely in Germany, and they have also been published in an English translation in "Traction and Transmission." One of the last works that the late Professor Thurston did was to abstract these experiments as a chapter in the new edition of his work on "Friction." Mr. Lasche plotted all his results in curves in three dimensions, and he found that the influence of temperature changes was very marked. I think that every one should take into account these important experiments in considering this subject.

The President.—It was my good fortune some twenty years ago to assist Mr. Hiram F. Mills, the distinguished engineer of Lawrence and Lowell, Mass., in some experiments in regard to the friction of various metals on one another in which this question of the presence of oil, that has just been touched upon briefly, was found to be far more important than is generally appreciated. It was found that apparently the early experimenters on friction had failed in the accuracy of their results from not appreciating how difficult it is to get rid of a film of oil on any piece of metal that has been machine worked. It is a matter for regret that the results of these experiments on friction by Mr. Mills, like many of his researches, have not yet been published. It was found that even the minute lubrication or oiling imparted by merely the passing of the hand over a metallic surface that had been carefully cleaned and freed from oil by chemical means or by cutting to a fresh surface apart from the presence of oil, would very materially change the friction when measured with precision, and that the co-efficients of friction as laid down in the text-books for surfaces free from oil were very materially in error, apparently from the very minute quantities of oil that were present on the metals that had been subjected to the experiments.

Any one who has worked at the process of lithography realizes how difficult it is to keep the surface of a clean lithographic stone absolutely free from oil. Merely the placing of a recently washed human hand on a clean lithographic stone will leave enough oil on its surface so that this impression can be "worked up" and "brought out" so as to give a distinct print of the finger marks. That illustrates how very difficult it is to keep all oil from a piece

of metal that has been handled. I offer the suggestion to any one who is experimenting on friction in a precise way, that it is an exceedingly difficult matter to be sure of the absence of oil on the surfaces and requires exceeding great care to be certain that you are getting metal to metal with no infinitesimal film of oil between.

*Prof. Price.**—I desire to make acknowledgment to those who have discussed the paper, and to thank them for the kindly attitude which they have assumed in regard to it.

The criticism offered by Mr. Moss concerns the manner of lubricating the bearings. Possibly a more complete flooding with oil would give more dependable results. However, the progress of my own thought on this point is given in paragraphs 27 and 28 of the paper; to which I have nothing to add at present. Upon analysis it is seen that here the foundation thought was of the elimination of abrasion without destroying the opportunity for the metal to exhibit whatever speed-friction characteristic of which it might be possessed. Finally, the effects of accidental abrasion, foreign particles, etc., were palliated by the taking of a large mass of data in repetition.

* Author's Closure, under the Rules.

No. 1089.*

SMOKE AND ITS ABATEMENT.†

BY CHAS. H. BENJAMIN, CLEVELAND, O.

(Member of the Society.)

1. The subject of this paper is one which was suggested by the Secretary of this Society during the past year, and is, or ought to be, of general interest at the present time. The smoke nuisance has long been one of the bugbears of living in the middle West and is now rapidly becoming an important factor in the East. It is so much easier to keep this nuisance out than to get rid of it when once in, that the residents of cities on the Atlantic seaboard ought to be alive to the situation.

2. The fact might as well be recognized at once that the supply of so-called smokeless fuels, anthracite coal, petroleum, natural gas, etc., is limited and will not begin to satisfy the demand. The supply of bituminous coal on the other hand is practically unlimited, and it is doubtless the fuel of the future. The abundance of it, its cheapness and the readiness with which it burns, even with a poor draft, combine to make it a most desirable fuel. This granted, the further fact remains that it will be burned to a larger and larger extent in the East as well as in the West and that no legislation can prevent this. Just as anthracite coal becomes more scarce and its price increases, just so surely will its place be taken by the cheaper and more abundant fuel.

3. The problem for our engineers and lawmakers is not how to prevent the burning of soft coal, but how to burn it in such

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* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Vol. XXVI. of the *Transactions*.

† For further discussion on this general subject consult *Transactions* as follows:

No. 222, vol. 7, p. 788: "Smoke-preventing Furnaces under Steam Boilers." Topical Discussions.

No 968, vol. 24, p. 315: "Smoke Consumption." Topical Discussions.

a way that the combustion shall not be a public nuisance. Unfortunately this, our best fuel, is at the same time our dirtiest fuel both in the handling and in the burning. The smoke from most other fuels, while not suitable for the lungs of human beings, is, if carried to a sufficient height by a chimney, comparatively inoffensive, since it does not return to earth.

4. The present discussion will then be confined to the problem of abating the smoke from bituminous coal. The word abatement is chosen with a reason and the terms smoke prevention and smoke burning are discarded. It is not practicable to prevent the evil entirely, but only to mitigate it in a degree. Smoke burning on the other hand is an impossibility under the conditions which usually present themselves.

5. During my somewhat strenuous experience as city smoke official I encountered two extreme types of unbelievers. One was the private citizen who did not believe that manufacturers had any rights in the premises and who clamored for instant prosecution. The other was the manufacturer who considered that smoke was an index of prosperity and regarded any criticism or interference as malicious persecution. Between these two extremes were all sorts and conditions of men, some belligerent, some indifferent, some interested, but nearly all destitute of any appreciation of the true nature of smoke or the means of its abatement. I would not jeopardize my standing in this Society by insinuating that there are any such present.

6. It will be well at the outset to state three propositions: *a*. The smoke from bituminous coal is a nuisance, especially in large cities. *b*. Such smoke can in the majority of cases be easily abated. *c*. Such abatement can be made a source of profit to the owner of the plant as well as to the community. There need be no difficulty in establishing these propositions by precept and by example.

7. Objectionable black smoke is due to the presence of hydrocarbons in the fuel and is produced as follows: The hydrogen and carbon compounds in the coal are driven off as gas by the heat at a comparatively low temperature, and may escape unburned. In this condition they would not constitute smoke in the common sense of the term. If heated to a sufficiently high temperature in the presence of air they burn with a yellow flame. If the air supply is insufficient, is poorly mixed with gas, or if the temperature is lowered in any way, combustion is checked and carbon is

deposited in the form of soot or carried off with the gas as smoke.

8. I cannot illustrate this better than by quoting from a paper read last year before the American Association for the Advancement of Science.*

The combustion of hydrocarbons seems to be always complete at first. If one watches the slow burning of a lump of cannel in the open grate he will see a whitish or yellowish vapor expelled from the coal by the gradual heat of the fire. This is the carbon and hydrogen combined which is distilled by the heat and leaves behind the free carbon as coke. While the escape of this vapor unburned represents a distinct loss of heat, the vapor is not smoke as we understand it. It does not deposit soot and will not stain or disfigure surfaces in its path.

As the heat increases, and air is supplied, the vapor ignites and burns with a yellow flame showing the presence of solid particles. If the temperature remains high and the air supply continues, the combustion is complete and the colorless carbon dioxide and water vapor pass up the chimney. If, however, the burning gas becomes chilled by contact with the relatively cool bricks of the chimney back or if insufficient air is supplied, the yellow flame becomes red and dingy, while particles of finely divided carbon are deposited on the adjacent surfaces or whirled away up the chimney.

The ordinary coal-oil lamp is one of the best illustrations of perfect combustion and consequent smoke prevention. The heated gases rising in the chimney produce a draft and fresh air is continually drawn in at the bottom through the hot gauze which warms and divides it so as to insure thorough mixing with the gases from the burning oil. Turn up the wick and the flame becomes smoky—too much hydrocarbon for the air supply. Raise the chimney slightly from the bottom and again there is smoke—too much air at too low a temperature which chills the flame. Insert a cool metal rod into the chimney and soot is deposited on it—chilling of the flame again and disengagement of the carbon, while the hydrogen continues to burn.

And thus we may learn of the three requisites for good combustion; enough air, a sustained high temperature and a thorough mixing of the gases. The last two are so important that it is entirely possible to have an excessive supply of air and dense black smoke at the same time.

9. It is difficult to form any estimate of the amount of damage inflicted by soft coal smoke in a city like Pittsburg or Cleveland, but it probably amounts to hundreds of thousands of dollars annually. The deterioration of dry-goods, draperies and delicate fabrics on the shelves and counters and in the show-windows is a direct tax on the buyer. In the book trade the situation is perhaps worse. I remember seeing in a display window in Chicago an assortment of valuable books, most of them open to view and all of them so streaked and spotted with soot as to render them unsalable at any decent price. Our libraries, public and private, are suffering day by day, and our art collections grow more and

* *Science*, March 25, 1904

more dingy. Cleveland can produce an old master "while you wait."

10. One of the largest items of annual expense in the city hotel or club-house is that of cleaning draperies, rugs and furnishings, and of redecorating walls and ceilings. A great part of this is directly chargeable to coal smoke.

11. The presence of more or less sulphur in the soot renders it a corrosive as well as a discoloring agent and greatly increases the damage done. Trees and shrubs suffer in such an atmosphere and are frequently killed outright by the presence of sulphurous smoke. The effect on human beings has not been definitely determined, but the deposition of soot on the delicate tissues of the respiratory organs can hardly be beneficial.

12. In law a nuisance is "such a use of property or such a course of conduct as, irrespective of actual trespass against others or of malicious or actual criminal intent, transgresses the just restrictions upon use or conduct which the proximity of other persons or property in civilized communities imposes upon what would otherwise be rightful freedom." (Century Dictionary.) Smoke is then a nuisance and can be prohibited by State or municipal enactment as the case may be. I have no more right to deluge my neighbor's premises with soot than I have to empty my garbage can over the fence line.

13. The abatement of smoke is in principle extremely simple, but presents some difficulties in the practical application. Only three conditions are necessary for complete combustion, the proper temperature, the proper air supply, a thorough mixing of the air and the hydrocarbons. The last condition is as important as any and is one too often neglected. It is this condition which gives the gas or liquid hydrocarbon an advantage over the solid, since the atomizing of the former by the steam or air jet insures the most intimate contact between the air and the fuel.

14. The use of pulverized coal in combination with air or steam is a close approximation to the above, and, when properly managed, gives good combustion, no smoke and a high efficiency. The cost of pulverizing and the impracticability of storing pulverized fuel have so far hindered the more general adoption of this process, except for metallurgical work.

15. When coal in the ordinary form is used as a fuel, smoke abatement involves some means of varying the coal supply and the air supply according to the demands made upon the boiler.

When ordinary hand firing is resorted to, the great irregularity of the coal supply will cause poor combustion and smoke unless the air supply is varied to correspond. Steam jets are frequently employed under these circumstances, and, if properly put in, will improve the combustion by drawing in additional air over the grate and mixing it with the products of combustion in front of the bridge wall. The steam jet should be semi-automatic, the steam and air being turned on by the opening of the fire door and gradually closed off by a dash-pot attachment.

16. The best solution of the smoke problem, so far, has come from the introduction of mechanical means of handling the coal, which give a uniform feed to the fuel and a corresponding delivery of air for combustion.

17. The use of mechanical stokers has been brought about by the natural demand for machine handling in large power plants as more economical than human labor, rather than by a philanthropic desire to benefit the community.

18. It has been estimated that one able-bodied man with a shovel and slice-bar can take care of 200 horse-power of boilers. With good mechanical stokers he can handle double, and with complete coal and ash-handling equipment three times this amount.

19. Stokers may be divided into three principal classes: Inclined, shaking grates—traveling or chain grates—and underfeed stokers.

20. The inclined grate, as exemplified in the Wilkinson, Brightman and Roney stokers, has a hopper in front and slopes down and back, having a clinker grate just in front of the bridge wall, while the double incline, as in the Murphy and Detroit stokers, has a magazine on either side and slopes in two planes parallel to the axis of the boiler, meeting in a clinker grate at the bottom. The principle of action is practically the same and involves the slow coking of the coal on a dead-plate, the pushing forward on to the top of the incline and the gradual descent, impelled by oscillation of the grate bars, until the combustion has left nothing but ash and clinker at the bottom. Air is usually admitted both below and above the grate, and the hydrocarbons which are distilled at the top of the grates pass through the intense heat of the burning coke on their way to the bridge wall and are completely burned. The double incline usually has a revolving clinker bar which disposes of some of the ash automatically, but as a rule

both forms need considerable cleaning. When used with a fuel which does not cake or clinker too much and when not crowded too hard these stokers are economical and reduce the smoke considerably. If, however, it becomes necessary to slice and poke the fire on account of caking coal or overcrowded boilers, unburned masses of coal are rolled to the bottom and holes are made in the fire through which cold air rushes. Both of these circumstances make for poor combustion and a smoky fire. As a rule firemen poke the fire on stokers too much, doing more harm than good.

21. I have seen inclined grate stokers, carrying a heavy fire and developing much more than the rated capacity of the boiler with very little slicing and hardly a trace of smoke. Cleaning is usually a source of black smoke for a few minutes, but this is mostly unnecessary if the fireman understands his business and gets his fire ready for cleaning beforehand. I think the middle door in the Murphy furnace is sometimes a disadvantage, as it tempts the fireman to meddle with the fire when he had much better leave it alone.

22. The traveling or chain grate is rapidly coming into favor as a means of stoking automatically. It consists, as in the Babcock & Wilcox and Green stokers, of an endless horizontal chain running on sprocket wheels and carrying the coal back under the boiler, finally dumping the refuse over the back and into the ash-pit. The distilling process and the gradual burning of the coke are much the same as in the stokers just described. In order to prevent waste through the grates the latter are usually quite close, and it becomes necessary to use more draft than with ordinary grates.

23. A damper is used underneath the grate to prevent an excess of air from passing up behind the grate. Some tests recently made by a large corporation which uses a considerable number of the chain grates showed an evaporation of only 5.7 pounds of water per pound of coal. An examination disclosed the fact that a large excess of air was passing through the comparatively bare grate at the rear end. The introduction of a damper to regulate this brought about a great improvement. With the same fuel and same conditions as before an evaporation of 8 pounds of water per pound of coal was obtained. Intending users of chain grates would do well to bear this in mind.

24. From observations covering a period of several years I have come to the conclusion that this type of grate is the best one yet

devised for abating smoke. The fact that it is horizontal, so that unburned coal cannot run to the rear end, and the further fact that it is self-cleaning and need not be disturbed by the slice-bar, make it an almost ideal furnace in this respect. If run by an intelligent fireman who understands adapting the depth and travel of the fire to the work to be done, it will also be a very economical furnace.

25. The underfeed stokers operate on an entirely different principle, the coal being fed in underneath the grate and forced up through a rectangular opening in the center. A forced blast is used and the air for combustion is blown up through the coal, the tuyeres being on either side of the rectangular opening just mentioned. By this arrangement the fresh coal is always underneath and the distilled gases are obliged to pass through an incandescent mass of fuel in company with the air. With a proper pressure of blast perfect combustion is thus almost unavoidable. The ash and clinker are now at the top of the fuel, which forms a gradually rising mound in the center and pushes the clinker over to either side, whence it is removed by hooks through doors at the front. The heat generated is such that the ash generally melts and forms a sheet of clinker which can be easily removed without disturbing the fire.

26. In the American stoker the coal is forced under the grate and up by a revolving screw, somewhat similar in shape to the ordinary gimlet pointed lag screw.

27. In the Jones underfeed stoker a plunger driven by steam operates to feed the coal. This plunger can be arranged to start and stop by hand or to run automatically.

28. My experience has shown the underfeed stoker to be economical in operation and practically smokeless.

29. A criticism frequently made in regard to mechanical stokers is that they will not respond quickly to sudden changes in the load, that it is difficult to keep a uniform steam-pressure under such circumstances, and that for this reason they are not economical. There is some truth in this. It is easy to conceive of circumstances, especially in electric plants, under which it would be difficult to maintain a uniform steam pressure with either the oscillating or the traveling grate. For regular fluctuation of load, as in electric lighting or railway power-houses, the obvious remedy is the introduction of storage batteries and the provision for ample boiler reserve.

30. Minor fluctuations can be taken care of by the fireman un-

less they become too numerous or too violent. In the latter case the underfeed stoker with the plunger feed comes the nearest to satisfying the demand. With the power of instantly regulating the motion of the plunger and the pressure of the blast, it is thus possible to meet emergencies of this kind more promptly than by hand firing.

31. With the complaint sometimes made that stokers cannot be forced I have no sympathy. With an ordinary inclined grate stoker under a horizontal tubular boiler I have forced a boiler to seventy-five per cent. above its rating with practically no smoke and with an evaporation of over 8 pounds of water per pound of bituminous slack. It all depends upon the draft and upon the intelligence of the fireman. Probably, however, none of us believe in forcing a boiler to this extent; it is bad for the boiler, bad for the stoker and bad for the coal pile.

32. The economy resulting from the abatement of smoke is naturally a potent argument in its favor. This phase of the subject has, however, but little to do with the ethics of the question. Once prove that it is practicable to abate smoke and the community has a right to insist on its abatement regardless of economical considerations. It may cost me more to haul away my garbage than to throw it into my neighbor's yard, but that is no argument for my imposing upon my neighbor.

33. Fortunately this difficulty does not exist in most cases, for it may be stated as a general proposition that smoke abatement means economy in fuel consumption. The proof of this statement is extremely simple: fuel economy results from good combustion, good combustion is accompanied by little visible smoke. It must be remembered that the converse is not necessarily true, for a smokeless chimney does not always mean good economy. An excess of air may insure entire oxidation of the combustible matter and at the same time so dilute the chimney gases as to cause serious waste of heat.

34. The problem to be studied by the manufacturer should be how to obtain the most perfect combustion of the particular fuel which he uses and then the smoke question will take care of itself. Let him make frequent analyses of chimney gases, compare different pressures of draft and different dispositions of dampers and he will be repaid for his trouble in more ways than one.

35. Personally I believe in the automatic stoker as the most economical solution of the problem. In forming this opinion I do

not rely upon expert tests for efficiency; there are so many variables entering into the question that it is difficult to make accurate comparisons in this way. As it is entirely possible to improve the efficiency of a furnace ten or fifteen per cent. by intelligent hand firing, there is much difficulty in determining the actual saving effected by a stoker. During a competitive test between the hand-fired furnace and the stoker the conditions are often entirely different from those obtaining in every-day use, and these changed conditions are usually more to the advantage of the hand firing. By this I mean that good hand firing is the exception rather than the rule, on account of the dirty, disagreeable nature of the work and the low grade of help employed.

36. With the introduction of mechanical handling there is a reduction in the quantity of manual labor and there should be an improvement in its quality. Manufacturers should understand that if they expect to get the benefit of improved machinery they must have men competent to run it to the best advantage.

37. The surest method of determining the relative economy of the two methods is by comparing the coal bills before and after the change, making due allowance for any variation in the work done. This has been done in a number of instances to my knowledge and the result has always been favorable to the stoker. About a year ago I addressed letters to a number of business men and manufacturers who have installed mechanical stokers of various sorts in factories and office buildings, asking for information as to the value as smoke-abating devices of the particular stokers used, and as to the fuel economy and the cost of repairs.

38. The replies in every case were favorable to the stoker, both as regards smoke abatement and as regards general efficiency. There are in the city of Cleveland fifteen large office buildings, ranging from eight to sixteen stories in height, which have mechanical stokers in connection with their power and heating plants. It is doubtful if any such building will be erected in Cleveland in the future without an equipment of this sort.

39. One manager of a ten-story office building writes that he has three 200-horse-power stokers which have been in use for four years burning slack coal, and that the only repairs have been the replacing of brick linings at a total expense for the four years of only fifty dollars per furnace.

40. The following quotation from a letter written by the manager of a large storage warehouse explains itself:

"We installed under our four boilers stokers known as the ———, and they all went into service on March 1, 1903.

"Our chief object in installing the stokers was to abate the smoke, and this they accomplish to such an extent that our chimneys are practically smokeless, and in addition to this we saved 500 tons of coal for the year. This we think is a fair showing, and thus far we have been put to very little expense in keeping them in repair."

41. An extract from another letter is as follows: The stokers cost approximately $1,500. The annual saving of fuel is $550. The demand for steam is increased ten to fifteen per cent."

42. Personal letters from well-known business men in Cleveland testify to a saving in fuel varying from fifteen to twenty-five per cent. A very conservative letter from the chief engineer of one of the largest power-houses in the city contains some criticisms of faults in the various stokers used, but concludes as follows: "While it is impossible for me to ignore the weaknesses in the above stokers, yet I would certainly recommend the use of either of these stokers over hand-firing in a plant of any size using bituminous coal."

43. The business policy of some of the largest corporations forbids letters of commendation for any mechanical device whatsoever, but I am in position to know that exhaustive tests covering a period of months have shown the economy of the new way in a most satisfactory manner. This is further evidenced by the fact that these "soulless corporations" are changing to the mechanical stoking as rapidly as the pressure of business will allow.

44. One suburb of Cleveland containing large steel works has been redeemed in the most remarkable manner by the introduction of automatic stokers, so that there is now a place for human beings to live, where formerly all was soot and blackness. The president of a manufacturing company using four stokers reports that he is not only paying from fifty to seventy-five cents less per ton for his fuel, but is burning six tons less per day than formerly, and at the same time getting better satisfaction in the maintaining of a uniform steam pressure.

45. It is undoubtedly true that the agents of smoke-abating devices of all sorts have been extravagant in their claims and that they have damaged their own business by such claims, but this is a mistake not confined to the furnace business.

46. A conservative claim of from ten to twenty per cent. sav-

ing, depending upon how smoky the furnace has been, can usually be substantiated by a comparison of the coal bills before and after introducing stokers, without resorting to expert tests. I have known so simple a thing as an automatic steam and air jet to pay from 50 to 100 per cent. annually on the original investment, as determined by the coal bills.

47. For the benefit of those who rely largely upon expert tests for their opinions, I introduce a summary of several such tests which have come under my personal observation.

No.	Kind of Boiler.	Kind of Stoker.	Boiler H.-P.		Kind of Coal.	Per Cent. of Ash.	Evaporation from and at 212°	
			Rated.	Developed.			Per Lb. Dry Coal.	Per Lb. Combustible.
1	Return Tubular	Murphy	1670	2241	Slack	20.0	7.78	9.71
2	" "	"	1670	2046	"	12.9	8.88	10.14
3	" "	"	2087	1923	"	13.3	8.07	9.31
4	" "	"	175	232	"	19.0	9.75	12.
5	" "	Jones Underfeed	100	216	"	14.6	9.07	10.62
6	" "	Hand Firing	100	193	"	13.7	6.71	7.78
7	Scotch	Jones Underfeed	....	232	"	8.67	7.68	8.42
8	"	Hand Firing	....	170	Run of Mine	14.31	6.86	7.99
9	B. and W.	Chain Grate	600	852	Slack	13.1	10.08	11.60
10	" " "	" "	100	115	"	19.4	6.39	7.94
11	Return Tubular	" "	125	179	"	17.5	10.35	12.56
12	" "	Hand Firing	125	134	Run of Mine	12.	7.8	8.76

Tests 1 and 2 show the improvement made by more careful handling of the stokers, while 2 and 3 illustrate the advantage of overloading rather than underloading the plant.

In test 2 only twelve boilers were used and in test 3 the number was increased to fifteen, the load being nearly the same. The poor showing in test 1 was due principally to waste of coal in the ash.

Tests 5 and 6 show the improved results from stokers as compared with hand firing under a heavy overload, while Nos. 7 and 8 give the results of similar tests on Scotch boilers. No. 9 was an acceptance test for a city plant, conducted by trustworthy and experienced engineers.

No. 10 is introduced to show how not to do it. Not only is the ash high, showing poor judgment in running the stoker, but actual measurement proved the air supply to be excessive, about 350 cubic feet per pound of coal burned.

With induced draft such as was used in this test, it should be

possible to get along with from 200 to 250 cubic feet of air per pound of coal. Tests 11 and 12 show the possibility of carrying an overload with a chain grate, using a poor quality of coal and getting better results than could be obtained with a better quality of coal hand fired.

48. If I have said little in this paper about means for smoke abatement other than stokers, it is because I believe that the stoker is the best remedy where it can be used, and that it should always be adopted in new power plants.

49. Whenever for any sufficient reason the stoker can not be introduced, there are other devices which will mitigate the smoke nuisance considerably and also save fuel. The steam and air blast has already been mentioned. A recent improvement which promises well is the combination of steam jets at the bridge wall with oil vapor, creating an intense heat at that point and consuming the hydrocarbons as they pass through. The expenditure of oil is comparatively small and considerable economy is said to result.

50. Brick arches and baffle walls have also assisted in maintaining a high temperature and in properly mixing the gases. The use of a reverberatory furnace or outside oven in which the coal and its gases are thoroughly burned before going to the boiler is another satisfactory method of reducing smoke, and may be used as well with as without a stoker.

51. In short, whatever produces good combustion abates smoke. In apartment houses and stores where the boilers are used for heating only and the steam pressure is low, the use of fuels which are comparatively smokeless is about the only satisfactory solution. If the use of such fuels is supplemented by intelligent and careful firing, the results will be reasonably satisfactory.

52. Locomotive engines present an entirely new set of conditions requiring treatment radically different from that accorded to stationary plants. Several mechanical stokers have been designed for use on locomotives, and one of these at least has been fairly successful. This application is still in the experimental stage, however, and it is too early to pronounce upon its future.

53. Most railroad companies have to rely entirely upon the skill of their firemen for abating the smoke nuisance within city limits and some of them have made a great improvement in this respect.

54. Regular and careful observations during the years 1900, 1901, on the railroads entering the city of Cleveland, showed a re-

duction in the average amount of smoke emitted by locomotives of from one-half to two-thirds, as a result of the so-called one-shovel system of firing, without any change in the furnace or equipment. The introduction of brick arches also proved to be a good investment.

55. I quote herewith the instructions issued by the management of one trunk line entering Cleveland, during the year 1900. These correspond in the main to a set of rules printed in the "Railway Age," in July, 1899, which were originally adopted by the C. & O. & T. R. R. as a means of reducing the fuel consumption.

INSTRUCTIONS FOR FIRING LOCOMOTIVES ON THE —— R. R. TO PREVENT SMOKE.

Explanatory:

Bituminous coal contains a large percentage of hydrocarbons which are nearly all driven off as gas by the heat of the furnace in a few seconds after the coal enters the firebox. Hydrocarbons, when unconsumed, make objectionable black smoke, but when a sufficient quantity of air is mixed with the hydrocarbons, combustion is complete, and no smoke appears.

Rules:

After firing each shovelful of coal, the door must be left open one or two inches for a few seconds, admitting enough air to produce complete combustion of the gases driven off from the coal. Care must be taken not to leave the door open longer than necessary to consume the gases. Firemen must learn to work with as light a fire as possible.

Before starting, the blower must be put on and a sufficient supply of coal put in the firebox to insure a good, solid fire. After the coal has been put in the door must be left partly open, by placing the latch on the first notch of the catch, so to remain until the smoke entirely disappears, when the door must be closed.

The engineer should so arrange the water supply that the fireman may be able to fire the engine regularly and economically, and this can be done best when the water is supplied to the boiler continuously.

Firemen must pay particular attention to the manner in which the engineer works the injector and handles the engine, in order to regulate the fire accordingly. Care must be taken to have the blower applied and the door partly open when approaching a station at which a stop is to be made, and no smoke must be allowed to show from the stack at such times, or when descending grades.

While the blower is being used, except when approaching a station at which a stop is to be made, care should be taken to keep the door closed as much as possible, more especially when cleaning the fire, as the blower causes the cold air to be drawn into the flues. While lying on the side tracks, both dampers should be closed to save the fire.

Grates should be shaken only when absolutely necessary, as too frequent shaking causes a loss of fuel, by allowing the unconsumed coal to fall into the ashpan, where it ignites and causes the pan to heat and warp. Ashpans should be examined as frequently as stops will permit, and under no circumstances must they be allowed to become filled.

When possible to avoid it, fire-doors must not be left wide open. To leave the door wide open is especially bad when using steam or blower.

56. In closing I wish to call attention briefly to the legal aspect of the question, although this may not interest engineers as much as does the scientific aspect.

57. I believe that the municipalities have a right to insist upon the abatement of black smoke by all users of steam boilers, without regard to the purposes for which the steam is used or the means to be adopted for abatement. This, because smoke is a public nuisance and because it can be abated without hardship to the owner of the plant. Nevertheless, when the evil is present and has been present for a period of years, it is not good policy to be too radical in the enforcement of the statutes. The law should be definite and stringent and the penalties adequate, but they should be enforced with discretion by officials who have some technical and practical knowledge of smoke abatement.

58. It is absurd to talk of putting this matter into the hands of the police or of the health officer. The official having charge of this work should be a trained engineer, if possible a technically educated man, and he should be entirely above graft in any of its disguises.

59. Some years ago I was indicating to the municipal committee of a commercial club in a Western city qualifications for a smoke inspector somewhat similar to those just mentioned, when the secretary of the club, turning to his associates, remarked solemnly: "Gentlemen, I do not see but what we will be obliged to get an angel for this position."

60. As a rule, very little attention should be paid to complaints, since they are usually inspired by prejudice and neighborhood jealousy rather than by any knowledge of the actual conditions. The inspector should be in a position to know the condition and characteristics of each chimney, and these should be determined by systematic observations made by a trained assistant. The observations should be made each five minutes and should cover all the time during which the chimney is visible.

61. To summarize the facts and principles of smoke abatement, I would repeat:

(1) Black smoke is a public nuisance and should be regulated by legal means.

(2) It is a result of imperfect combustion and can be largely abated by proper methods of stoking.

(3) Mechanical stokers offer the best means of accomplishing this result, in medium-sized or large plants.

(4) As smoke abatement is a result of better combustion, economy of fuel is the natural and obvious result.

62. To these I would like to add one more, a principle which is common to all reforms:

(5) An educated and intelligent public sentiment must be the moving and compelling force, without which mechanical devices and legal enactments will both fail.

63. When the public comes to realize its rights in this matter and that it is under no more obligation to submit to this nuisance than it is to endure bad drainage or filthy streets, laws will be enacted and enforced and the people will wonder that they so long submitted to this unreasonable imposition.

DISCUSSION.

Mr. A. Bement.—This paper, like many others which have been written, treats the problem in a general manner and fails to offer a definite solution for any particular case or condition. Every one of the schemes that is mentioned or recommended is in some cases producing smoke, and doing so with the best obtainable manipulation. The general problem as it presents itself at present with a large variety of inefficient apparatus, receiving more or less indifferent manipulation, is extremely complicated, and like all such problems, requires analysis directed toward the solution of some definite feature before a substantial beginning may be made. It is first necessary to realize the fact that, broadly speaking, there are no smokeless "furnaces" or "stokers" that are manufactured for sale; therefore, a well-intending purchaser may buy the best available apparatus which, with a certain superior grade of coal and careful manipulation, may produce a smokeless condition, but he may not be able to afford the cost of such superior fuel, or to command the required careful manipulation. Everyone of the devices and schemes described in the paper are in a large number of cases making serious smoke and doing so under the most favorable conditions of fuel and manipulation attainable in the given cases.

Nearly all efforts made in this matter are mostly the outgrowth of present practice which is itself unsound, and we are largely

endeavoring to materialize our present hopes, rather than direct efforts to an accordance with natural physical laws, and as the solution of a complex problem may be arrived at only by properly directed detailed treatment, it is well to attack it at the beginning rather than endeavor to work backwards.

Taking this view of the matter, let us see what can be done in a certain definite case. Therefore, I will take that of the water tube boiler with horizontal inclined tubes. This type is selected for two reasons; first, it is one which by all means, of itself, causes the largest smoke production; and second, it offers the best opportunity for application of smokeless apparatus in series with it.

If a thorough mixture of combustible and air could be effected at the grate or in the fuel bed, or if it could be attained a few inches above the fuel bed, then the desired result would be immediately secured, but as such ideal performance is unattainable, and if it was, must be largely disturbed when ashes are removed, something other than a plain grate should be used. This leads to the consideration of a stoker machine which, if fulfilling requirements, must perform two functions; first, continuous and uniform supply of fuel and consequent evolution of combustible, as well as similar removal of ash. Second, to effect a complete mixture of gases at the fire. It is apparent that this first requirement is unattainable with any form of sloping grate or underfeed device, or, in fact, with any stationary or semi-stationary form of grate; therefore, it is not necessary to determine whether these devices will fulfill the second function, that of complete mixture at the fire, especially so, because this may be determined with the next form of stoker to be examined, which is the chain grate, an apparatus which fulfills the first requirement in an ideal manner, but which, like all general forms of stokers, receives its fuel at one end of the grate, at which point evolution of the coal gas occurs in a narrow zone, and that the fulfillment of the second requirement, that of mixture of the gases, does not occur, may be readily observed at the chimney top. Thus, following the stoker throughout its various forms to its most perfect type, that of the chain grate, we find it finally to be a failure as far as complete solution is concerned, because it does not cause sufficient mixture of gases with the air which is present.

This final and important requirement which the stoker fails to fulfill, which, as before mentioned is that of effecting mixture of

the gases, may be secured by locating a furnace between the stoker and boiler of such capacity that there will be opportunity for mixture to occur before the boiler surface is reached. Such furnace may be easily produced by covering the tubes of the bottom row of the boiler with a refractory tile, which extends from the front to about five feet from the back end of the boiler. The presence of this tile furnace roof requires, of course, that the gases enter among the tubes at the back of the boiler. These final conclusions lead to an apparatus which is "smoke proof," if the furnace is of a length of from sixteen to eighteen feet, measuring from the point of entrance of the fuel to the exit, and this is the only form of apparatus which produces smokeless combustion of itself independent of its manipulation.

I am, of course, aware that chain grates, served with expensive washed or especially sized coal, or operated with a very thin fire, often produce but very little smoke, but the combination of apparatus which I have described will produce a smokeless result when using the fuel that the stoker is usually sold to burn, and does so with that thickness of fire which gives the best economic result in steam made per unit of fuel, and which at the same time also makes the most smoke *at the fire*.

It is apparent that such combination of apparatus as above described cannot be obtained as a unit, therefore, strictly speaking, a smokeless apparatus cannot be purchased to-day in the market as an entirety. The first example of this combination was produced some years ago by the Chicago Edison Co., by locating this form of furnace between water tube boilers and chain grate stokers, which were already at the time in service, with the result that the Heine Safety Boiler Co. adopted this form of tile for its boilers, and the practice of attaching this element to a furnace has been extended to other similar boilers by some different makers, so a purchaser may buy a boiler and its furnace element from one maker, and a chain grate stoker from another, and in this way obtain a smokeless apparatus.

The above is a case of the proper principles in their best application, and if the action at the fire produced by other forms of stoker or that from common grates can be made to equal that produced by the chain grate by proper manipulation, then the final result will be equal, but this introduces the uncertain and uncontrollable factor, that of the personal equation of the fireman. I do not wish to appear as discouraging careful effort on

the part of firemen or furnace operators, because, if engineers cannot produce good apparatus, and a fireman may get good results from inferior devices, he is a man from whom much is to be expected.

I think that Professor Benjamin overrates the lack of proper public sentiment favoring smoke prevention. There are many people who are doing as well as they can with the imperfect apparatus at their disposal, many of whom have displaced one form for another without deriving benefit thereby. Also others who have purchased the best that the market afforded, and which make serious smoke, and they cannot stop it. As a matter of fact, it is the practice of the engineers who design and select the apparatus that is not keeping up to the pace set by owners and their operatives.

An inference in paragraph 22 should be corrected, which is, that owing to the small opening in the chain grate, a strong draft is necessary. Now, a chain grate stoker cannot operate with a very strong suction draft on account of excessive air leakage through the coal hopper, and the usual limit is about 0.6 of an inch of water, and with such draft, coal containing about ten per cent. of ash and a like amount of moisture, has burned at the rate of sixty pounds per square foot of grate per hour. Such coal was necessarily of a very favorable size, but this shows how much air can flow through these small openings, especially as it was as much as 100 per cent. in excess. I have found that the amount of opening through different grates has but little effect on the amount of the coal burned as far as air is concerned, but that the condition of the fuel bed has an enormous influence.

Another matter that should be mentioned, is that chain grate fires may be made to respond to demands of regular and irregular peaks of load as found in electric generating stations, if the operation of the fires and stokers is properly managed.

Mr. John T. Hawkins.—Professor Benjamin says, paragraph 32, "The economy resulting from the abatement of smoke is naturally a potent argument in its favor," which he qualifies, paragraph 33, by the admission that "A smokeless chimney does not always mean good economy."

In paragraph 46 he says: "A conservative claim of from ten to twenty per cent. saving, depending upon how smoky the furnace has been, can usually be substantiated by a comparison of

the coal bills before and after introducing stokers, without resorting to expert tests."

In the latter connection it is to be regretted that, in paragraph 69, where the party quoted says: "Our chief object in installing stokers was to abate the smoke"—and that, "in addition to this we saved five hundred tons of coal a year," he does not give us the total consumption for the year, so that we might arrive at the percentage of saving.

Similarly, paragraph 41, the party writing saved fuel annually to the value of $550, but does not say what was the total value of coal consumed per annum.

About the year 1880 the writer, in conjunction with the late Pheron Skeel, conducted a quite exhaustive series of experiments, with a specially prepared boiler setting, having for their object to determine, 1st, the possibility of burning bituminous coal of the various grades, in a boiler furnace, practically without producing smoke, and 2nd, what was the actual loss due to the production of smoke only, other conditions being equal.

Without taking time and space to describe this boiler setting, suffice it to say that it was such as to permit of very gradual and practically continuous firing by hand; with equally good means of regulating the air supply.

It was found entirely practicable to continuously burn either of the brands of bituminous coals to be had at the time absolutely without smoke.

It was also found possible, by varying the method of firing and the air supply, to produce at will any degree of smokiness desired at the top of the chimney.

One of the objects being to determine the value of unconsumed or uncombined carbon constituting the smoke proper, the results from the two methods of firing—smoky and smokeless—were carefully compared; and it was found that as between the most smoky and absolutely smokeless firing the loss never exceeded 0.7 of one per cent.

This would make it appear that the claim, paragraph 46, of ten to twenty per cent. saving could scarcely depend merely upon how smoky the furnace had been, but upon some other more important considerations, somewhere to the extent of nine to nineteen per cent.

It seems to me therefore that the statement that "the econ-

omy resulting from the abatement of smoke is naturally a potent argument in its favor, is somewhat misleading. If it is really as potent an argument as is indicated in paragraph 46, very few considerable consumers of bituminous fuel would be found with smoky chimneys.

Most of us older engineers will remember when traveling years ago at night on Hudson river steamers, burning anthracite coal in their furnaces, the more or less entertaining sight of a voluminous and continuous blue flame at the tops of the chimneys, which we, now at least, know came from the incomplete combustion in the furnaces, allowing large volumes of CO to take up at this point the additional atom of O necessary to perfect the combustion, and establishing a secondary furnace, as it were, at the chimney top, while all this should have been accomplished below. Doubtless in furnaces burning bituminous coal similar conditions may obtain, regardless of smoke, which alone never reaches a loss of one per cent. of the fuel.

I am quite at one with Prof. Benjamin otherwise, generally, and with paragraph 6 (c), except that the mere abatement of smoke can be any material source of profit to the owner of the plant, however profitable it may be to the community.

If it were true that the mere abatement of smoke could result in any such saving as one-fifth of the fuel, believe me, there would be no such smoky cities as Pittsburg is to-day.

There is no doubt, however, but that any contrivance or system which shall effect a perfect combustion of bituminous coal, or produce the most economical results, must at the same time effect the abatement of smoke; nor is there any question that mechanical stokers are now among the best means to that end; and certainly no one can at this late day fail to be convinced that for such large plants as admit of the application of mechanical stokers, a smoky chimney has no excuse. If the Vienna engineers could so effectively avoid smoky furnaces as is described by Mr. Gus. C. Henning (*Transactions*, Vol. 24, p. 323), we American engineers should not balk at such a problem.

Aside, however, from such extensive plants as permit of or warrant the use of mechanical stokers, there are a thousand and one smaller plants of an infinite variety, down to the heating apparatus of private houses, which offer an inviting field for the inventor in some contrivance, arrangement or attachment which shall ensure smokeless furnaces under the kind of management

which is necessarily inseparable from such cases. For as Professor Benjamin well says (paragraph 1): "Just as anthracite coal becomes more scarce and its price increases, just so surely will its place be taken by the cheaper and more abundant fuel."

Mr. Geo. H. Barrus.—Professor Benjamin has had considerable experience with this subject, being a city official whose duty it is to endeavor to abate the smoke evil so far as the law will allow. Having unusual opportunity for studying and observing the effect of smoke-preventing devices, his remarks and opinions are entitled to much consideration.

In my own observations connected with this problem, I find that steam users are willing enough to adopt measures for preventing smoke, but they object to going to much expense in the matter, and especially when it comes to the installation of automatic stokers. Such an innovation as the attachment of a stoker often calls for the entire rebuilding of the furnaces and iron boiler fronts, which is an undertaking that not only interferes with the operation of the plant while the change is going on, but it involves a large expenditure of money. The cost of a stoker is sometimes one-half the cost of the boiler itself, and steam users object to putting out this amount of money merely for the prevention of smoke. Even under compulsion of the statute, few can be prevailed upon to adopt this method.

Although it is true that the automatic stoker, as Professor Benjamin so well states, is an excellent device for preventing smoke, it is equally true that it is such an expensive mode of accomplishing the result there is little use in recommending it for universal adoption. In view of this aspect of the matter it seems to me that Professor Benjamin has reached an impracticable conclusion, and I would like to know how he proposes to meet such a serious objection.

I notice in paragraph 49 the mention of a combination oil burning appliance in connection with a coal furnace for reducing the quantity of smoke. I should like to know more about this. Perhaps the Professor will make a fuller explanation.

In paragraph 18 there is a remarkable statement that one able-bodied man can take care of 200 horse-power of boilers. I am sure this cannot be taken seriously. I have in mind a number of boiler plants where one fireman handles three, four, and even five times this amount of boiler power.

Professor F. R. Hutton.—The secretary would state that the

experiments conducted by Mr. Barrus were with anthracite and not bituminous coal.

Mr. A. J. Herschmann.—I happen to be in a position to be able to say something about the use of bituminous fuel in New York City. The Flat Iron Building, as is perhaps known, is one of the very few buildings in New York City that uses soft coal under its boilers. We have had no difficulty with smoke, and it has been our experience that with a careful fireman no black smoke was generated. Recently the Hawley furnaces in the Fuller Building were removed, and we introduced a system of blowing air under the fire, combined with automatic control of the damper. At first I looked at it in a very sceptical way, for I did not believe that with a very high stack it would be possible to increase the economy of a furnace by introducing mechanical draft. However, when the device was introduced, we found an increase in economy. We used mechanical draft under the fire, burning soft coal, and we found there was hardly any smoke emitted. The chief reason is that with the use of this device there was a perfect balance of draft above the fire and no rush of suspended carbon into the stack. Ordinarily with a high stack when the furnace door is opened there is an inrush of cold air which chills the blaze and causes a considerable amount of smoke at the top of the stack; whereas, with the use of the balanced draft as now applied in the Fuller Building, it is found that all the crevices, so to speak, of the boiler are filled by the gases which are given a thorough chance to mix.

I would like to say also that 200 horse-power, of course, is not a great amount of work for a fireman. We generated almost 450, and one man had no special difficulty in firing the boilers. Consequently, as regards stokers it is my opinion that a stoker could not be a source of great economy, particularly in a small place where you have to handle your coal anyway. It would be different in a large plant, a central station for example, where special coal handling is resorted to, but in an office building where the fireman is expected to handle his own coal—or at most one coal passer employed—I do not believe a stoker would add to the economy. As regards smoke affecting economy, we found that it did not. In one case when we found a great deal of smoke emitted, the economy varied only about one per cent. The most serious question with soft coal is the gas created by it in the boiler-room and the coal dust raised in handling it.

Mr. Ambrose Swasey.—As one of the guilty manufacturers of Cleveland who has done his part toward eclipsing the sun in that section of the country, I wish to speak of the excellent work done by Professor Benjamin during the time he was supervising engineer, and in charge of the smoke abatement for the city of Cleveland. I am glad to say that he took up the solution of this problem in a way which commanded the respect of every manufacturer and steam user in the city, and had he not been obliged to contend with the political methods of the city in addition to the smoke nuisance, I am sure that Cleveland would be a much cleaner city than it is to-day. He certainly did abate the smoke in Cleveland to the extent that the difference could readily be observed on the streets. He also built up a strong public sentiment in favor of smoke abatement, and I am confident that when such work is taken up and carried on in the way in which Professor Benjamin started it, a large amount of the smoke in our western cities can be done away with.

Something has been said in regard to automatic stokers and hand firing. Judging from my own experience I am satisfied that when automatic stokers are applied to boilers having sufficient capacity, the smoke can practically be eliminated. As to hand firing, we can do much more than we would at first believe. On one occasion I entered our boiler-room and found the fireman shoveling in a large amount of coal. I asked him why he put so much coal on one side of the boiler and none on the other, as there were two openings. His answer was that he thought it was the best plan to put a lot of coal on one side and when the proper time came, to throw in as much on the other. I suggested that he throw in small amounts of coal, approximating the automatic stoker as nearly as possible, and then told him that his position would depend upon the top of the smokestack, and from that time on the amount of smoke was greatly reduced. I think that excellent results may be obtained by hand firing, and that fully 75 per cent. of the smoke from the average boiler using soft coal can be avoided.

Mr. Max H. Miner.—There is something said in the paper about the one shovel method of firing locomotives. It seems to me that that is something that is going to require attention before long, unless electric locomotives take the place of steam locomotives in our cities. At the time when this one-shovel method was first agitated the size and power of the locomotives were

such that in most cases it was entirely practicable, but in the last five or six years the size of locomotives, the size of the boilers, and the power required has increased so much that when a locomotive is fired by the one-shovel method it is in most cases a practically continuous process. In a test made on the Erie, eleven coaches and Pullman cars were hauled up a heavy grade, from Port Jervis to Summit, in about half an hour, the locomotive developed continuously about 1,200 horse-power and the fireman inside of half an hour fired 150 shovels of coal, which would be on an average about five scoops a minute. Of course it was necessary to close the door between firings in order to lessen the amount of cold air going in, the draft being tremendous, and it was necessary for him to fire about six or seven scoops at a time. Every time he put in a scoop full of course there would be a cloud of smoke for a short time, but probably a large amount of that was due to the fine dust in the coal being drawn through and out of the stack. It seems to me that if some of the smoke nuisance on heavy locomotives is going to be abated by anything resembling the one-shovel method it will require something in the line of mechanical stokers which can fire the coal continuously in small amounts and yet keep the fire door closed, or closed with the exception of a small opening.

Mr. Paul R. Brooks.—I come from Chicago and I think—with all due deference to our past President, from Cleveland, that there cannot be much question of the demonstrated right of Chicago to a place in this discussion. The railroad and tug boat smoke nuisance there is at least as bad, if not worse, than any of the manufacturing plants, and I have seen something of both sides. One reason for the steamboat nuisance is that one man will be obliged to handle all the way up to 2,000 horse-power of boilers with soft coal, and he fires, as Mr. Swasey's man did, so that it will stay fired for a sufficient length of time for him to attend to the other boilers. The result is an almost continuous black fog. The railroad proposition has been solved in two ways. One of the companies tried to burn coke. They were not altogether successful. Yet with no change excepting a draft appliance in the front end of the locomotive and a partial mixture of coke and semi-anthracite coal they have been able to get along very nicely. Another railroad was unable to get anything to abate smoke from switching engines until they designed a type of locomotive with a very large fire box in comparison with the amount of heating surface, and

burning a moderate fire in that. With the switch engine firemen they found it very hard to get any result at all on the one-shovel firing plan. You know, the switch engine firemen are the poorest of this class of labor, and that was an obstinate source of trouble at first. But the solution in general has seemed to be in arranging for a rate of firing which approximates a steady supply of coal and which makes not too great a demand upon either the fireman or the grate surface in the matter of combustion. With care and the exercise of a little common sense, very satisfactory results can be obtained without expensive installations. The tug boat nuisance is still open to argument.

Mr. William H. Blauvelt.—Professor Benjamin in his fourth conclusion states: "As smoke results," etc. I would like to ask the experience of the society in regard to the economy of the different classes of boiler installations. I know of a large company which has three boiler plants, each of 3,500 horse-power, standing in a row. One of them is old fashioned, and the coal is delivered on a trestle and then wheeled or shoveled to the front of the boilers; the ashes are raked out on the floor in front of the boilers and then carted away. There are no economizers. In the second one the coal is delivered from a bin above and run down through chutes in front of the fire doors, convenient to the fireman's shovel; there are tunnels under the ash pit and economizers in the rear, and forced draft is used. In the third one there is everything required for a complete modern plant; the coal is delivered by conveyors into a bin overhead, and thence to stokers of a standard type; the ashes are taken out from a pit underneath into cars and run off by power; there are ample economizers and induced draft, and every labor saving appliance. Careful analysis of the costs show that it is impossible to distinguish between the three houses. Including all costs from the time the coal leaves the car until the steam is delivered, including coal, labor, repairs, and interest on the different investments, the totals are practically identical, although those three boiler houses may be said to represent three different systems. Anthracite is used throughout this plant. I was discussing the subject the other day with one of the officials of a large steel company, and he told me that his experience in Cleveland was practically the same with soft coal. There they put in automatic firing, but they did not find it was of any economy, when all costs of labor, repairs and interest on the investment were included. This morning I visited the collieries of the

D. L. & W. Company, and I made the same inquiry. I found one boiler house there where firing was done by hand and the ashes taken away by hand, and there are no economizers. I was told they are getting their steam for less money than other collieries nearby where the boiler houses are fitted with stokers and all other improvements. It seems to me this is a very important part of the subject, and I should like to know what the experience of the members has been as to the relative cost of operation, as well as the relative evaporative efficiency of the coal, under the different systems.

Mr. S. Balkwill, Jr.—My experience has been that when we get good combustion with a chain grate we do our firing smokelessly. We have attached to our boilers an automatic arrangement for controlling the damper, and we find that when the steam pressure rises suddenly, causing the dampers to be closed quickly, we get a diminished amount of draft which causes imperfect combustion and at such times we get a small amount of smoke, but at those times only.

Mr. Fred W. Taylor.—I have had personal experience with three types of automatic stokers. Having gone at the stoker problem with great enthusiasm, I was surprised to find that when part of the boilers in a boiler house were equipped with automatic stokers and others were fired by hand the economy in both cases was about the same. My conclusion is, after a close observation of three of the leading types of stokers, that if the price of coal is very high, as it is in New England, for instance, there is little, if any, economy in automatic stokers, but where the labor of stoking forms a large item of the expense, the mechanical stoker is a very distinct advantage.

Mr. Blauvelt.—Why, if the automatic stoker is an economical appliance, is that true with high priced coal? It would seem to me that the advantage in that case would be greater.

Mr. Taylor.—An experienced, skilful fireman using coal of the best quality can with care obtain a rather higher efficiency than is usually reached with automatic stokers. And this increase in efficiency is sufficient, when the price of coal is high, to pay for his wages. With cheap fuel of poor quality, on the other hand, the mechanical stoker will give about as good efficiency as hand firing. The cheaper the coal the greater becomes the item of firemen's wages as a factor of the total cost, so that in this case stokers are an economy.

Mr. A. S. Mann.—I wonder whether anybody here has observed the chimney of the Metropolitan Station at 96th St., New York City. I think the New York authorities arrest the engineer when they find the chimney smoking and I believe they have no trouble.

I had charge of that plant at one time and know that they take great pains and get good results. They have over 80 boilers, half of them burning soft coal, and conditions are as nearly perfect as they can be made.

Mr. Taylor.—In New England practically nothing but high priced fuel is used, so that the location of the boiler plant has much weight in determining whether stokers are economical or not.

Mr. Miner.—There have been some experiments by the Illinois Central Railroad in Chicago, burning pulverized coal on a small scale, and it seems to me it would be interesting if the society could get some definite reports of those tests. I know in a general way that they claim a gain in economy of about 16 or 20 per cent., and it is a fact that there appears practically no smoke.

Mr. John R. Freeman.—There is no paper on our present list whose subject matter concerns every good citizen more than that of this paper by Professor Benjamin, and it devolves particularly on the members of this society to conduct a campaign of education on this subject.

Not long ago I was going through a very large factory in New York City and in the back part of the boiler room I noticed several large crates of rubbish, and asked the superintendent the purpose of saving it and storing it. He explained that they were particular about the enforcement of the smoke ordinances and that they had found this material made less smoke when burned after dark.

*Prof. Chas. H. Benjamin.**—I should take a decided exception to Mr. Bement's statements in his opening paragraphs: "Every one of the devices and schemes described in the paper are in a large number of cases making serious smoke and doing so under the most favorable conditions of fuel and manipulation, etc., etc."

Whatever may be the results in Chicago, I know that in Cleveland the mechanical stokers mentioned in the paper are in nearly every case preventing objectionable smoke, and are doing it with bituminous slack or nut, the cheapest fuel in our market.

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* Author's Closure, under the Rules.

Furthermore, when the stokers do allow smoke, the result is in every case directly traceable either to crowding of the boilers beyond a reasonable limit, or to carelessness or indifference on the part of the firemen.

I make this statement, not offhand, but as a result of systematic observation of all the stacks in Cleveland for a period of two years by trained observers. The device that Mr. Bement describes is familiar to me and, when used in conjunction with a good mechanical stoker, is a help to perfect combustion.

In reply to Mr. Hawkins I would say that a horse is an animal, but an animal is not necessarily a horse. Good combustion means a clean chimney. The reverse is not always true. Perhaps I should have stated that the amount of carbon in soft-coal smoke rarely exceeds one-half of one per cent. The real losses are due to the escape of unburned gases, but they have black smoke for their danger signals. The letters read were quoted literally and I cannot give the total amount of coal burned in any case.

Mr. Hawkins says, "If it were true that the mere abatement of smoke could result in any such saving as one-fifth of the fuel, believe me, there would be no such smoky cities as Pittsburg to-day."

It is strange, but nevertheless true, that the fact of fuel economy in complete combustion does not result in an immediate revolution in boiler practice. No stranger, however, than that other sources of economy in modern city life are similarly neglected.

An appreciation of the economy resulting from the use of mechanical stokers has effected a great change in the prevailing practice in Cleveland, and a corresponding improvement in the atmosphere.

One member of this Society, Mr. Alex. E. Brown, has made a successful application of the reverberatory furnace in combination with chain grates, locating them in front of the boilers and getting complete combustion before the gases pass to the tubes.

Mr. Barrus' remarks are good and to the point, but apply rather to the Eastern practice in burning anthracite coal than to the conditions obtaining in the Middle West. The economy from burning successfully a cheaper grade of fuel has usually paid a good interest on the original cost of the stokers. I am entirely serious as to the 200 horse-power limit for one fireman when handling soft coal slack at the rate of 5 lbs. of coal per horse-power hour.

I would not ask a fireman to do more and I doubt if Mr. Barrus would, after trying it himself for ten hours.

It is not good economy to crowd either the boiler or the fireman too hard.

The use of steam and air jets has been alluded to. It is a very effective method of smoke prevention, and is used very largely in Cleveland. I have not given it very much weight in my paper because it depends so much on the individuality of the engineer and the fireman. I have found that men who used that means had to be watched so continually and reprimanded so much that I got out of patience with that sort of prevention. The divergence in opinion as to economy is due largely to a difference in the kind of coal used. My remarks apply to bituminous coal. Economy I regard as a secondary consideration. Of course, the manufacturer does not. But I believe we would have a right to prevent the emission of black smoke from chimneys in cities if it cost the manufacturers twice as much as it does now to run for furnaces. I do not think that has anything to do with the legal aspect of the case. In western and central cities I believe there is a considerable economy resulting from the use of stokers, because it enables the manufacturers to get an evaporation of seven or eight pounds of water per pound of coal from a fuel that will only give them five or six pounds with hand firing. That is with the very cheapest kind of fuel they can buy. This has been demonstrated so many times to me in the course of my experience that I have no hesitation in affirming my belief in the economy of the mechanical stoker for handling the kinds of fuel that are used in that part of the country. I should not think that there was much economy in using stokers for handling high-priced fuel. The question has been asked as to whether coal improves in quality as we go east. I should state that proposition in another way: I know it certainly deteriorates as we go west.

I want to express my appreciation of what Mr. Swasey has said with regard to the work in Cleveland. If all the manufacturers in Cleveland were like Mr. Swasey the post of smoke inspector there would be more or less of a sinecure.

Allusion has been made to what has been stated by an official of a certain steel corporation with regard to economy at their plant in Cleveland. If this corporation has got to the point where it is putting in mechanical stokers for the sake of improving the

atmosphere of the city of Cleveland, it has experienced a decided change of heart since I have had any dealings with it. I had a great deal to do with the officials of one steel corporation when I was a city official, for the reason that they made three or four times as much smoke as any other corporation in the city, and when I approached one of the high officials on the subject, his language was such that I do not wish to quote literally, but it was very much like that imputed to Mr. Vanderbilt upon one occasion with regard to his disposition towards the public. Furthermore it was stated to me by an official of the company that they were experimenting with chain grate stokers at certain of their plants and making elaborate tests, of which I was allowed to have copies, and that the future disposition of the corporation depended entirely upon the results of those tests as to economy in coal. Following that, their plants have been equipped from time to time with stokers, and these plants are now practically free from smoke. In other words, that part of Cleveland which was formerly an inferno has been rendered at least habitable, as is noticeable to any one who is acquainted with it.

I believe that in city limits the solution of the locomotive problem is going to be the use of the electric locomotive or the use of smokeless fuel. It is undoubtedly true that it is difficult, if not impossible, to reduce the smoke to a satisfactory minimum with the one-shovel system of firing, especially with the larger types of locomotive. I want to express my appreciation of the hearty co-operation of all the railroad lines in Cleveland in connection with this problem of smoke abatement. The railroad officials assisted me in every way in their power, and made an improvement which was noticeable for its contrast with what occurred in a great many of the stationary plants. Perhaps this was largely because the discipline on the railroads was so much better; they had so much better control over their engineers and firemen and looked after them so carefully, that it made the problem comparatively easy of solution.

In closing this discussion, I wish to say that I have not tried to point out a remedy for every set of conditions that might exist, but rather to state certain general principles.

I would have all people know that smoke is a nuisance, morally and legally; that it can in a large measure be prevented, and that economy to the manufacturer is entirely a secondary consideration

to the public at large. The recent decision of Judge Kohlsaat in Chicago hits the nail on the head.

I would see the time when the black smoke at the chimney top shall be an emblem of piracy as much as the black flag at the mast head, and when the question of the profitableness of piracy shall be relegated to its proper place.

No. 1090.*

CONTINUOUS MEASURING AND MIXING OF CRUSHED OR POWDERED MATERIALS IN ACCURATE PROPORTIONS.

BY E. N. TRUMP, SYRACUSE, N. Y.

(Member of the Society.)

1. Many chemical and metallurgical processes require the accurate proportioning of the raw materials in a crushed or powdered form, and their intimate association by mixture until the particles are near enough together to admit of the desired reactions.

2. The success of these processes largely depends on the accuracy of the proportions, as an excess of one material, or lack of another, may give irregular results.

3. Many products are simple mixtures of raw materials, and where the color or appearance must be uniform, the proportion must be perfect.

4. The methods of the apothecary are sufficient for the compounding of medicines which are weighed in small quantities, but when materials must be mixed by the ton these methods become too slow and laborious.

Methods of Measuring and Mixing.

5. The developing of chemical processes has been an enlargement of the chemist's methods, and we find the favorite plan the "batch method." The material is measured or weighed in convenient sized "batches" and brought together in some kind of a mixing machine to be mechanically thrust or tumbled about until the particles are supposed to find their proportional distance from each other. The smaller the batches are made the less time required for the mixture, but the multiplying of the operations

* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

becomes a very monotonous occupation for the men intrusted with the weighing or measuring, soon resulting in careless work and requiring constant checking and testing to keep up the standard of quality.

6. The use of Hopper Scales, or of Automatic Scales, has improved the proportions, but the materials are still delivered in batches of considerable size, and must be thoroughly mixed by an intermittent process corresponding in time with the weighing, and difficult to make automatic. Even when the mixture of a batch is good it is often difficult to discharge the product from the mixing machine without destroying the accuracy of the mix because a difference in the sizes of the particles, or a difference in their specific gravity, tends to re-distribute the ingredients in the receiving hopper, which must be large enough to hold the contents of the mixer.

7. Weighing Machines are affected by the varying condition of the materials, which stick to the hoppers, or the dust which clogs the delicate balances. The gates used to feed the weighing hopper, or discharge its contents, leak, and foreign substances clog or prevent their proper action.

8. Where the material contains varying amounts of moisture frequent tests must be made and the weights changed to obtain the proper portion of dry material.

9. The measuring of batches of material is subjected to the variations in the personal equation of the attendant. The height of fall into the measure, the temperature and condition of material, the method of striking off surplus, the amount of tamping the material receives, all affect the result.

Continuous Measuring and Mixing.

10. Many machines have been made, and are in use, to proportion materials and mix them continuously. It is recognized that much saving of labor in handling, and greatly enlarged outputs, will result from proper methods.

11. The first element of continuous mixing is a continuous feed, which will be rapid and accurate, and not affected by varying conditions of the materials, or at least adjustable to these conditions. The following methods may be mentioned:

Screw Conveyors: Are used for discharging a stream from a hopper, the quantity being regulated by a sliding gate at the end or by changes in speed.

Shaking Spouts: Operated under a hopper to move the material delivered upon it.

Revolving Pockets: Receiving material from a spout and emptying into a spout below.

Revolving Rolls: Arranged to feed a measured quantity by the distance between them, or from a spout by extending the natural slope.

Reciprocating Tables: Moving horizontally under a spout, and moving the material out over its edge.

Revolving Tables: With stationary spout grinding material out until it falls over its edge, or is diverted by an inclined blade.

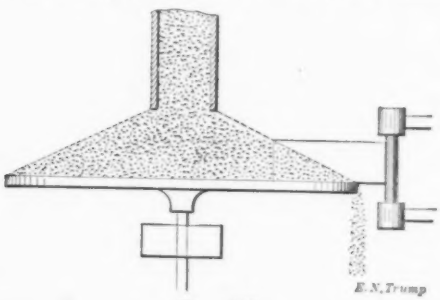

FIG. 252.

12. Each one of the above methods are in use in various forms, with different degrees of success. Some of them work well on a particular material, but most of them are affected by changes of speed, pressure from the hoppers holding supplies of material, changes of moisture, or size of crushing. Nearly all of them depend for accuracy on the natural slope assumed by the material as it discharges from a spout, and as the natural slope which the material assumes in falling out of a spout is changed by any variation in pressure from above, size of material, content of moisture, and speed of delivery, all of these affect the accuracy of the delivery.

13. Of the methods mentioned above, the revolving table, with a stationary spout above its center, as shown in Fig. 252, is much used in cement manufacture for feeding mills, etc. Its defect is the change in the natural slope of the material, with different pressures in the feeding spout which varies the amount cut off by the diverting blade.

14. If we make the table of relatively large size and distribute the material in a uniform layer upon it, as shown in Fig. 253, the

stationary knife, or diverting blade, may be pivoted so as to take off a predetermined quantity in one revolution, and if the layer is thin, variations in the natural slope on the edge will affect the accuracy very little. As the table revolves the stationary knife will divert the material in front of it over the edge of the table and it will fall in a continuous stream into the chute.

15. If the material is replenished on the table so that the layer taken off by the knife is restored to exactly the same shape as before, and is continuously removed by the knife, an accurately measured quantity will be diverted.

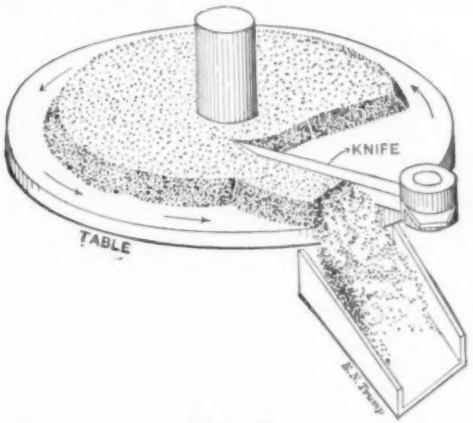

FIG. 253.

16. By adding to the table a bottomless storage cylinder, somewhat smaller in diameter, with its lower edge spaced a distance above the table sufficient to clear the knife and yet near enough to the table so that the material flows out from under the edge of the cylinder and takes its natural slope, we shall have the condition represented by Fig. 254.

17. The cylinder being supported by arms from the central spindle may be filled to the top by means of the chute, and as the knife removes the section represented by its path over the table, the material from the cylinder above will take the place left vacant, and will come out under the edge of the cylinder to the extent of its natural slope. While this slope may vary a little this variation is a very small part of the amount diverted by the knife, and as the material composing nearly the whole base of the cylinder is cut away the space behind the knife is filled from above with nearly uniform pressure, and in practice the natural

slope angle is almost exactly the same, in spite of considerable differences in height of material within the cylinder.

18. After deciding on the distance between the bottom of the cylinder and the table, and the width of the knife, the other factors, which determine the amount of material measured off in a given time, are the speed of rotation and the depth of the cut of the knife, and these are both adjustable. The depth of cut of the knife is adjusted by swinging the knife around on its pivot

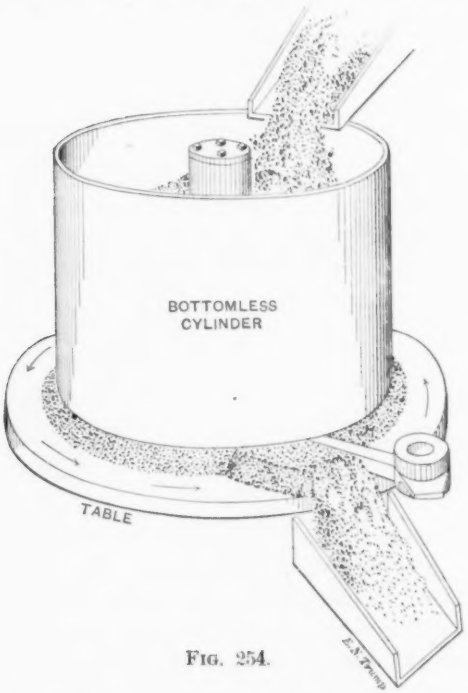

FIG. 254.

so that it extends a greater or less distance into the material. This swing is controlled by a screw attached to an arm, cast as part of the knife, and a micrometer scale with pointer shows the amount of movement. This is shown in Fig. 255.

19. The mechanism described above may be employed for the feeding of a great variety of materials, varying considerably in size and consistency, and if the size of the table, the shape of the cylinder, and the size of the knife and space between the cylinder and the table are properly adjusted, almost any kind of crushed material may be fed.

20. The variations in size may extend from fine powders, like cement, to rocks of 6 inches cube. In the case of the larger sizes the space between the bottom of the cylinder and the table is made considerably higher than the height of the knife, as the space between the bottom of the cylinder and the top of the knife must be sufficient to let the largest pieces pass through without catching. The amount diverted by the knife is not dependent upon its height, but on the height of the space under the cylinder.

21. As an example, in the case of a table 12 feet in diameter, feeding rocks of 6 inches cube, the variations for a single revolution, where some 6,000 pounds were diverted, showed less than 2.5 per cent., and for 10 revolutions the variations were less than $\frac{1}{2}$ of 1 per cent. of the quantities. With finely powdered materials the variations in the quantities by volume are considerably less than this.

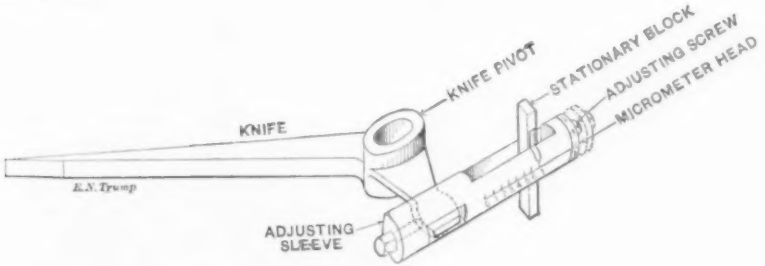

FIG. 255.

22. A measuring table of 3 feet 6 inches in diameter is being used as a meter measuring the quantity of powdered material delivered to a series of furnaces. The quantity was formerly ascertained by passing the material through a hopper scale, where batches were continuously weighed by an attendant. The measuring table was calibrated in different positions of the knife, and the total quantity measured by it was checked for two consecutive months. The variations from day to day, where 12 to 15 tons of material were measured, were less than 50 pounds, and the total differences for a month, for a measurement of 400 tons, showed in one case 108 pounds, and in the other case 150 pounds. As these variations were both sides of the average, the results of the scales were as doubtful as those of the measuring machine.

23. The same machine filled to the top of the cylinder was revolved and the material for each revolution weighed separately, the depth of the cylinder being about 3 feet, the quantities for

each revolution were within $\frac{1}{10}$ of a pound of each other, on a weight of 5 pounds, and no variation occurred until the material reached within 6 inches of the bottom of the cylinder, when it began to decrease a little in quantity for each revolution.

24. Having now obtained a method of measuring the materials, the proportioning of two or more materials, and the subsequent

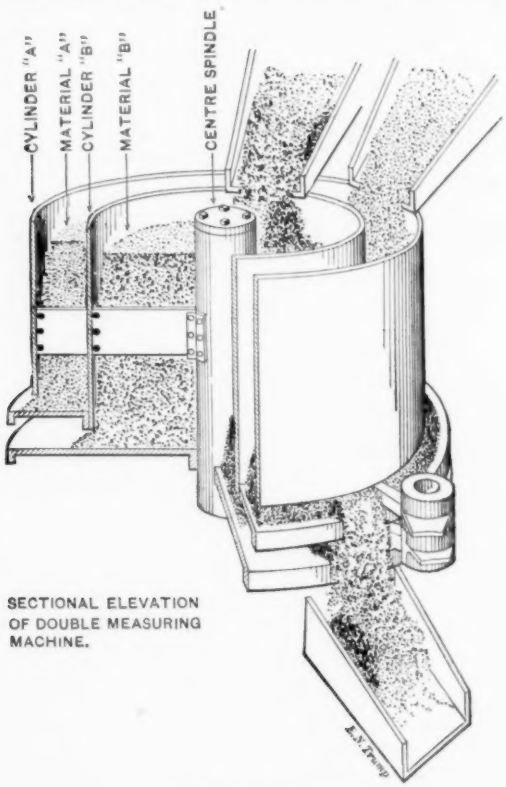

FIG. 256.

mixing, becomes a matter of the combination of two or more cylinders. Two measuring machines, such as have been described, may deliver into the same spout, and may be driven at different speeds, or the same speed, and by adjusting the knives any proportions may be secured.

25. As the function of a perfect mixer is to deliver the materials in the proper proportions in as close contact as possible, it is important to bring them together in superposed sheets while

falling from the table, so that little subsequent mixing will be required. To reduce the amount of mechanism, therefore, and have only one revolving part, the arrangement shown in Fig. 256 is adopted.

26. The two or more tables are set one above another, and the cylinders supported from the same spindle, so that they all revolve together. Each table has its own storage cylinder, or annular space between the cylinders, and for each table there is a knife with its own adjusting mechanism. These knives may be adjusted at will to vary the percentage of material in a mixture, and when the proportions are approximately known, the width of the knives and the distance between the cylinders and the tables are fixed at approximately the average desired.

27. As the material is diverted over the edge of the upper table it falls into the material coming from the knife next below, and these two quantities fall into the material from the third table, in case three tables are used.

28. The spouts which feed the storage cylinders may be so arranged that the revolution of the cylinder automatically keeps the cylinder full, the material itself preventing more issuing from the spout than is necessary to maintain a constant level in the cylinder.

29. Hoppers may also be employed, into which the material can be dumped from wheelbarrows, fed with shovels, or drawn intermittently from a spout at comparatively irregular intervals, the only attention required being to keep the cylinders reasonably full. The volume may be allowed to vary 50 per cent. without interfering with the accuracy of the quantity diverted by the knife.

30. With very finely divided powdered material, which flows very freely, it may be necessary to place a special mechanism in the top of the storage cylinders to regulate the pressure from the contents of a large bin so that it will not force the material to flow with too great a difference of velocity, these finely powdered materials being prone to stick in the bin, but when once in motion to have almost the mobility of water.

31. The design of this measuring machine may be considerably modified for different uses. One form of the machine is shown in Fig. 257 in which three tables are used, the tables and storage cylinders being all mounted and driven by the same center sleeve, which is carried on a dirt-proof step-bearing on top of a stationary spindle.

32. The lowest table has a bevel gear cast on the under side, which is driven by a pinion mounted on a countershaft. A ring projecting down from the outer edge of this table runs in a dust seal in the bottom casting. The casings of the machines are made dust-proof, to avoid any possibility of leakage, and the whole is mounted on a substantial base. The three knives can be readily adjusted by the micrometer adjusting screws shown on the right, and a closed delivery chute and receiving chutes may be readily connected to the cover and delivery spout.

33. Figs. 258, 259, 260 and 261 show the arrangement of single and double machines of this character.

34. Figs. 262 and 263 show the same machines for use with very finely powdered materials, having a special sealing device to prevent pressure from the entering spouts.

35. These machines may be used for the measuring and mixing of various chemicals; measuring and mixing of dry colors; for mixing of glass batch; for the proportioning of ingredients for the manufacture of special plasters; also of materials for making sand-lime brick; the blending of coffees; the measuring and mixing of flour, sugar, etc., in bakeries and confectionery establishments; for the combining of ingredients in fertilizers, and for a variety of chemical and industrial processes in which accuracy of measuring and mixing and reduction of labor, and especially of skilled labor, is desirable.

36. As a feeding mechanism, in continuous furnace processes for mills, or crushers, requiring regular feeds, and especially for the rotary kilns used in cement making, these machines have been found to be especially well adapted, and to materially increase the efficiency of the mechanism on account of the greater regularity of the operation.

37. One adaptation, which gives promise of great use in the industrial arts, is the proportioning of the cement, sand and stone, with the addition of the necessary water, in the manufacture of concrete. The strength of concrete depends largely on the perfect coating of the fine sand with the proper quantity of cement, and the proper filling of the voids between the crushed stone. If the proportions of the ingredients vary considerably some parts of the structure will be weak, and to guard against this weakness larger quantities of cement are necessary than would be the case if absolute accuracy of proportions was obtained.

38. The usual methods of proportioning the materials in con-

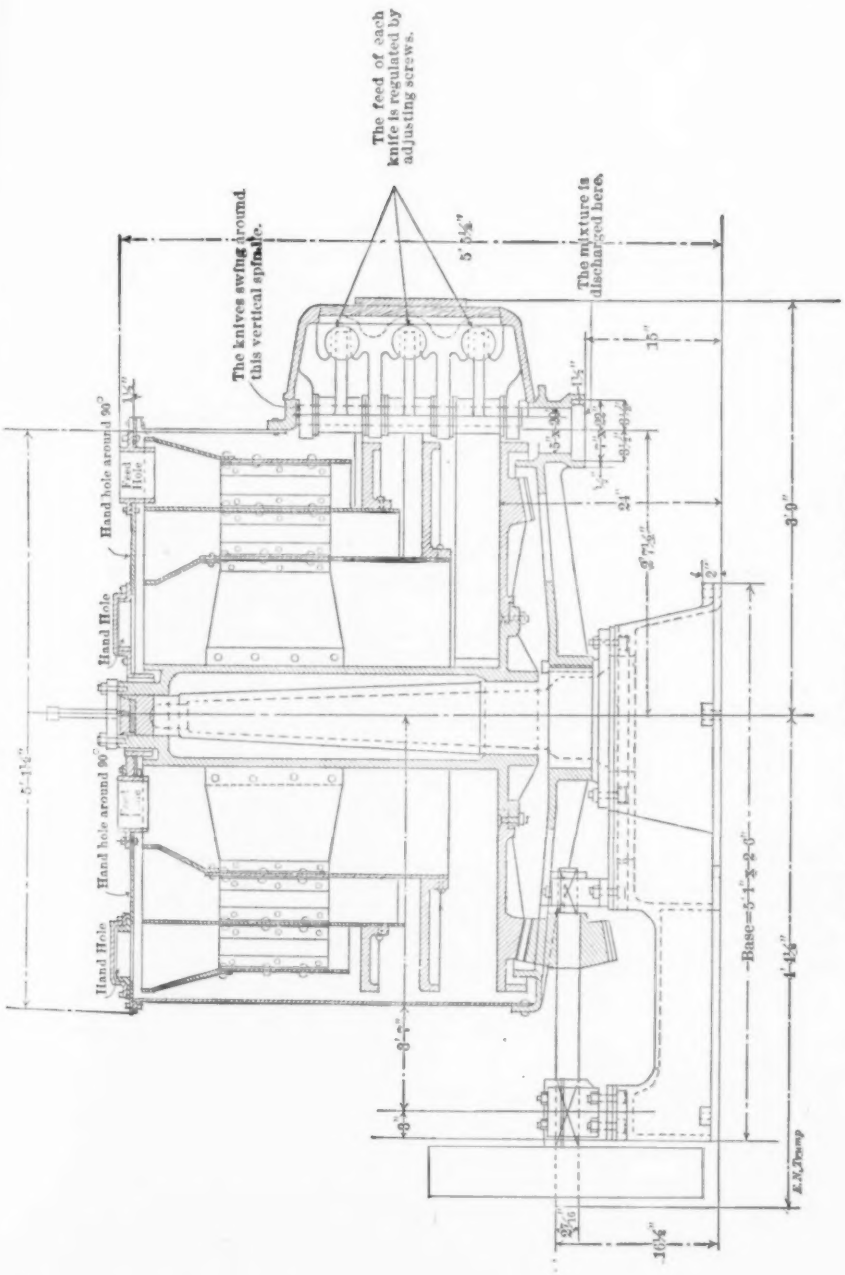

FIG. 257.—60-INCH TRUMP TRIPLE-MEASURING MACHINE.

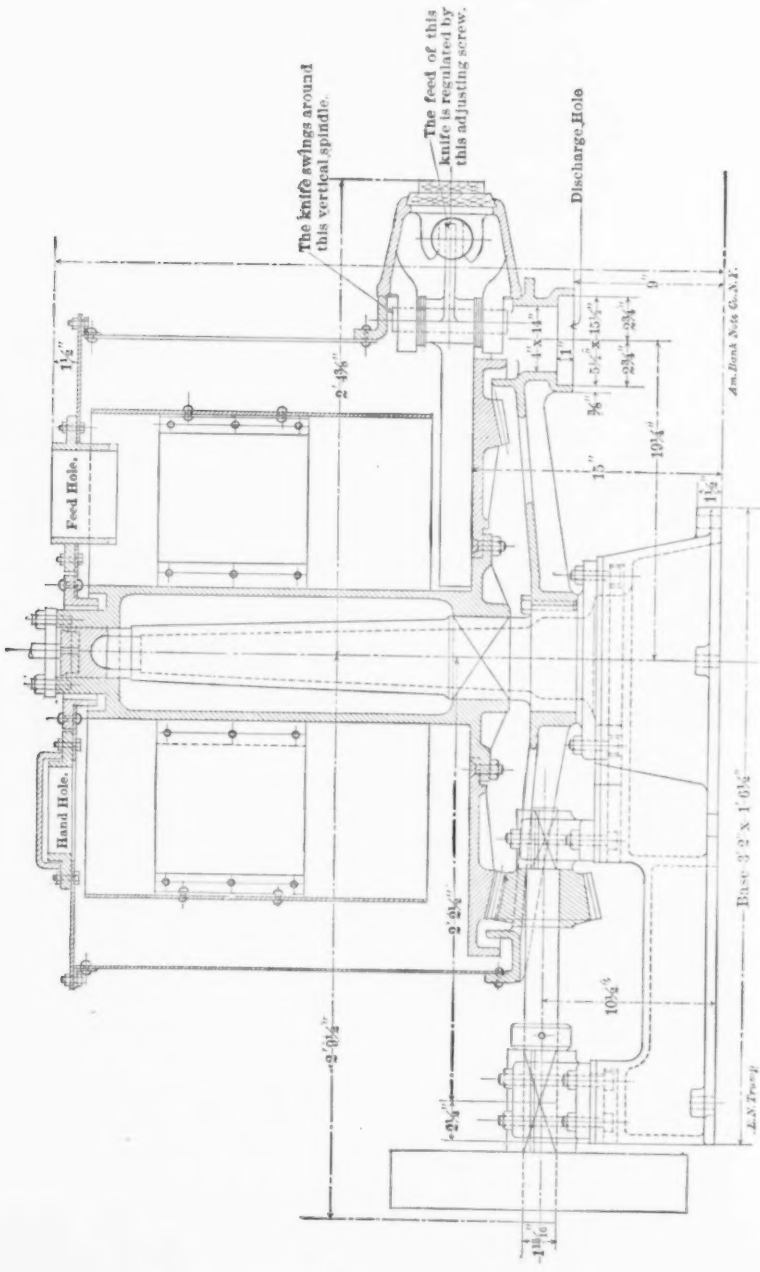
The knife swings around this vertical spindle.
The feed of this knife is regulated by this adjusting screw.
Discharge Hole
Feed Hole.
Hand Hole.
Am. Book Mfg. Co. N.Y.
E.N. Trump
Base 3'2" x 1'6½"

FIG. 258.—36-INCH TRUMP SINGLE-MEASURING MACHINE.

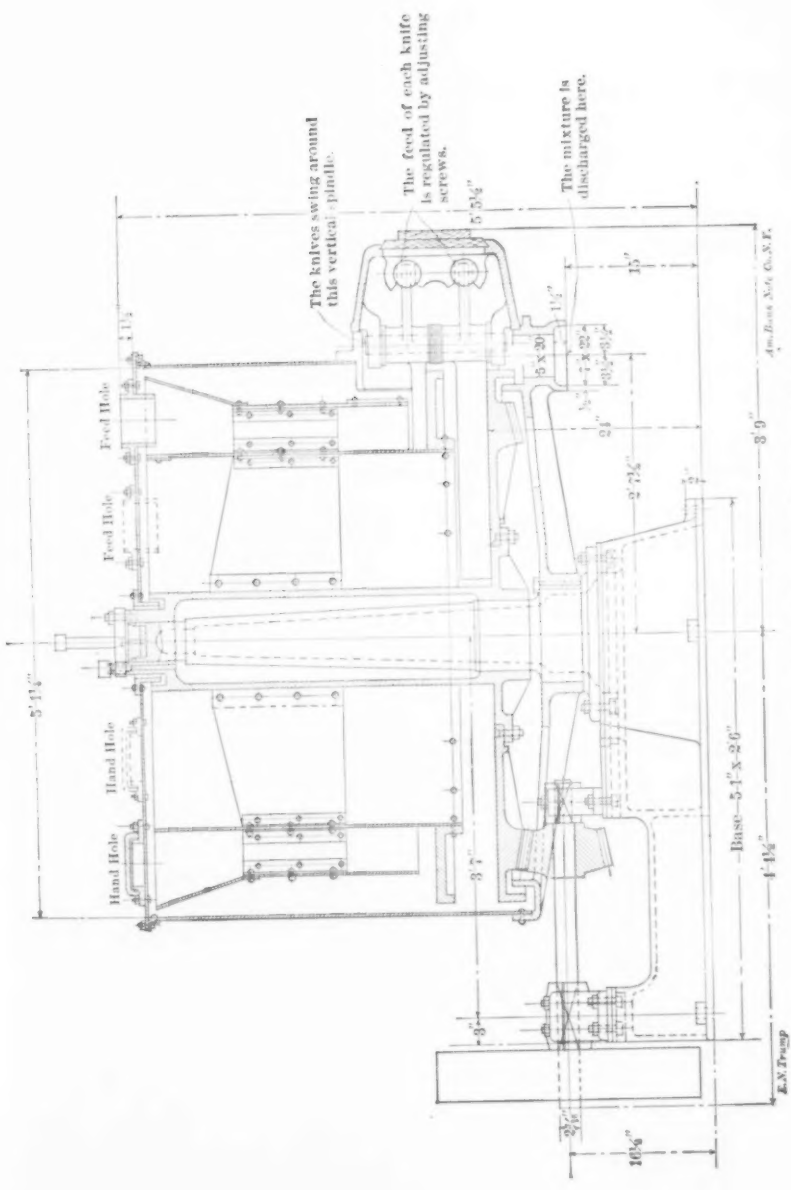

FIG. 259.—60-INCH TRUMP DOUBLE-MEASURING MACHINE.

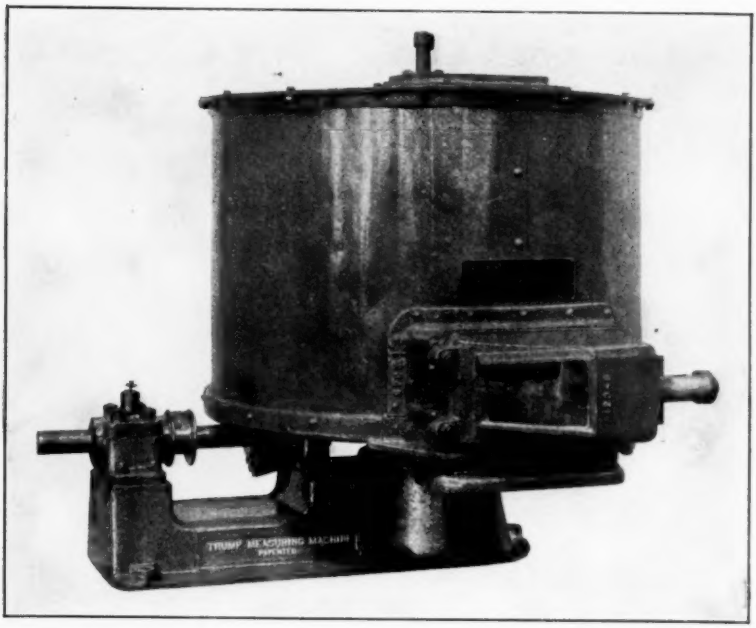

FIG. 260.

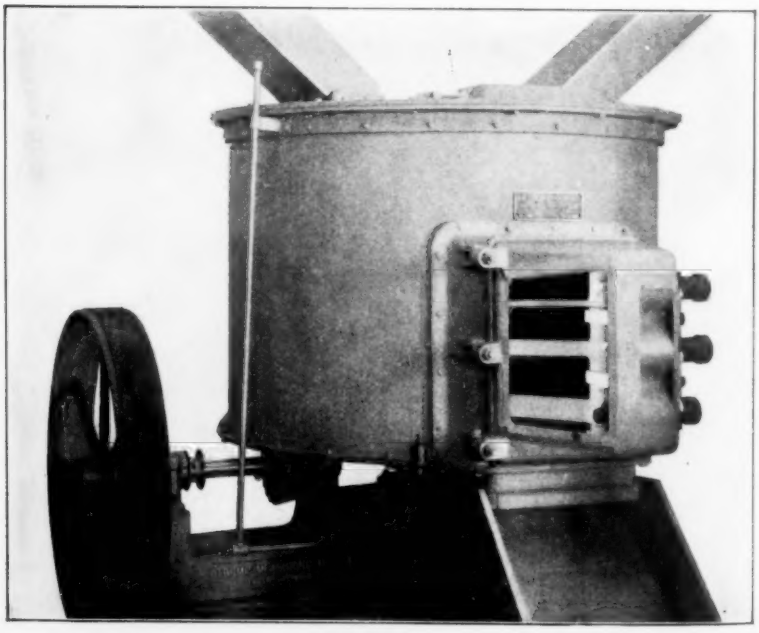

FIG. 261.

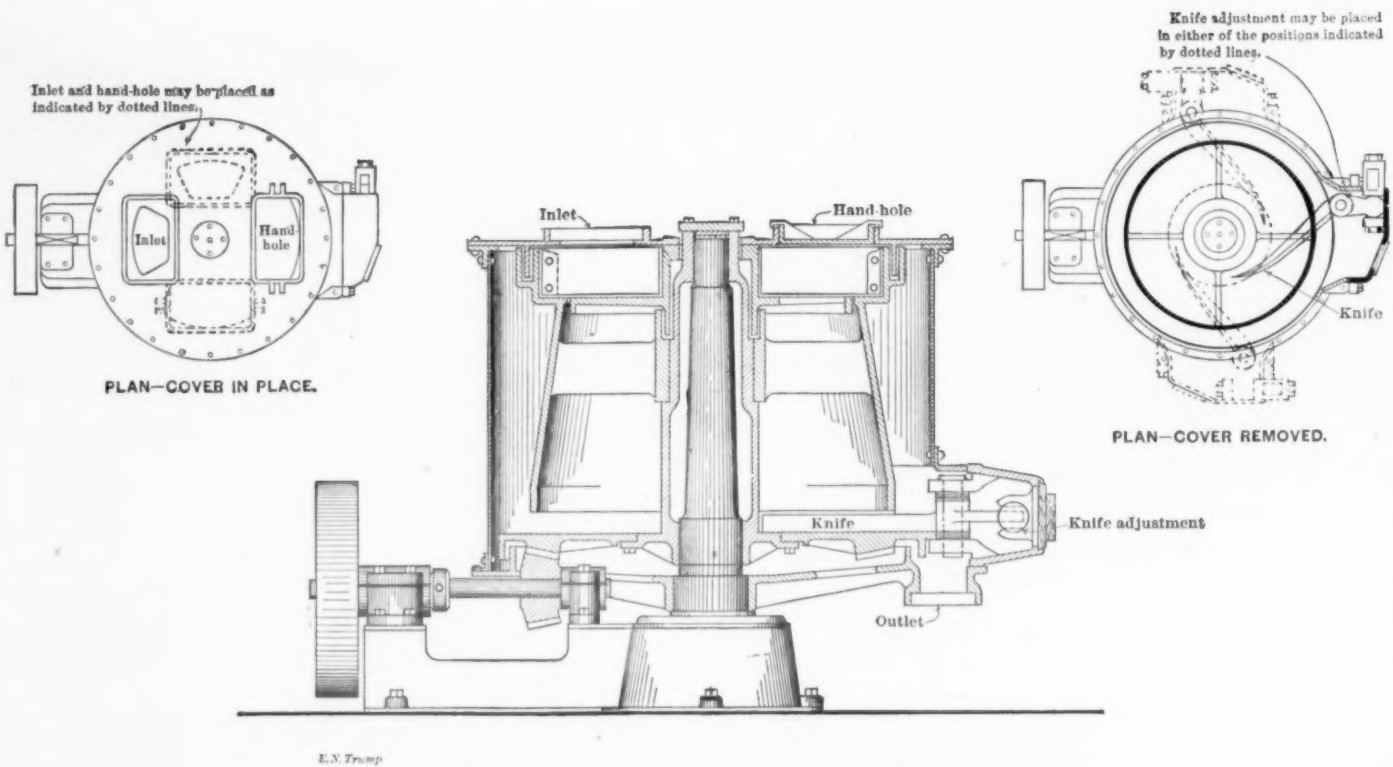

FIG. 262.—36-INCH TRUMP SINGLE-MEASURING MACHINE FOR FINE MATERIALS.

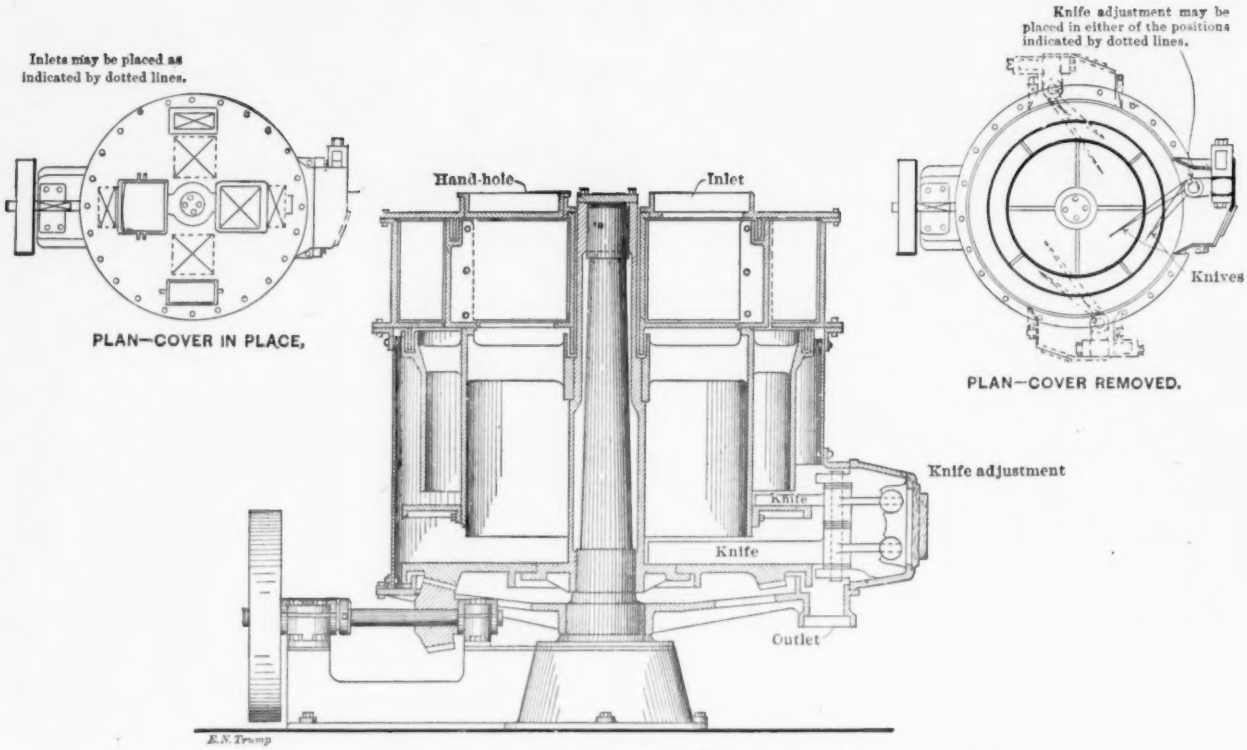

FIG. 263.—36-INCH TRUMP DOUBLE-MEASURING MACHINE FOR FINE MATERIALS.

crete mixing are very crude, and engineers are beginning to realize that for the important work now being done with reinforced concrete, greater accuracy is necessary. The method of measuring the ingredients in wheelbarrows, or by means of shovels in the hands of the ordinary laboring help, is very unsatisfactory, and the effort to make up for this deficiency, by using machine mixers, does not entirely remedy the trouble.

39. Figs. 264 and 265 show an arrangement of measuring machines, adapted to concrete mixing, which has been very success-

FIG. 264.

ful. Fig. 264 is intended for a semi-portable machine, having a capacity of 50 cubic yards of concrete per hour, the materials being fed from temporary stationary hoppers supplied with ordinary elevating mechanisms, and delivered by the measuring machine through a double spray of water into a short interrupted screw, so that the concrete is delivered in a continuous stream accurately proportioned and thoroughly incorporated with the water. With this machine about 10 horse-power is required for measuring and mixing at the rate of 50 cubic yards per hour, and as small a quantity as a cubic foot can be made with the same accuracy as a much larger quantity.

The diverting knives of the measuring machine can be quickly changed to suit the proportions required in the concrete, and are

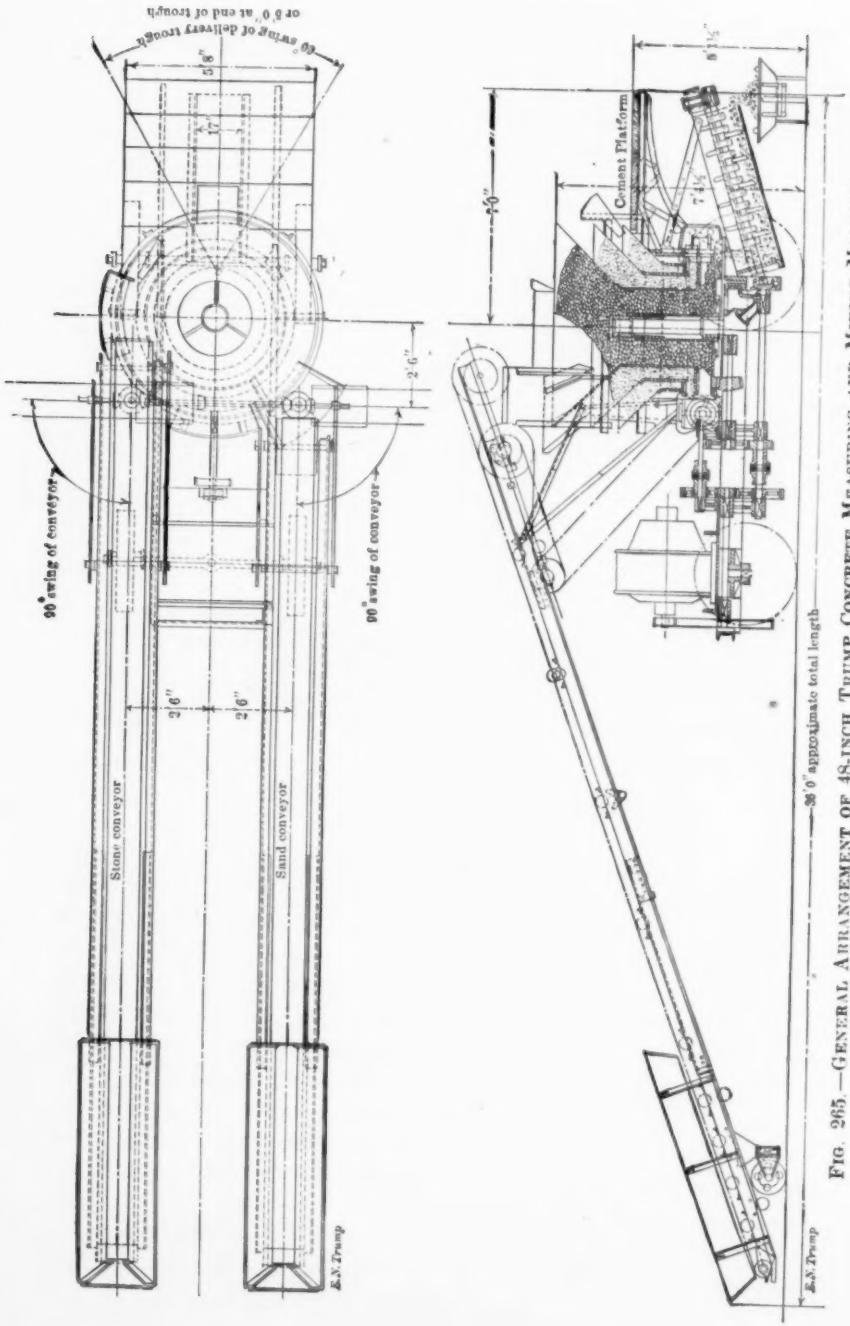

FIG. 265.—GENERAL ARRANGEMENT OF 48-INCH TRUMP CONCRETE MEASURING AND MIXING MACHINE.

provided with a locking arrangement so that they can be set by the superintendent and not changed except by his direction.

40. Fig. 265 shows a more portable arrangement of the same style of machine, designed to have a capacity of about 30 cubic yards per hour, in which special belt conveyors are provided for receiving the sand and stone at the level of the ground on each side of the machine, the cement being delivered to a platform and fed into the machine from bags by an attendant. These machines may be driven by a motor or engine and boiler, and the interrupted screw mixer has vertical and horizontal movements, which are useful in delivering to wheelbarrows, into excavations, or onto the street level in street paving work.

41. The great saving effected by the accuracy and continuity maintained by these machines have been proven in actual practice, and the accuracy demonstrated by use in actual construction. The only limit in speed is the ability to get material into them fast enough, and as only the interrupted screw conveyor contains any mixed concrete no cleaning is required, and the machine can be stopped and started with little delay.

Experiments on Single Measuring Table, 4 ft. 6 in. Diameter.

42. Comparisons of the amount of soda ash fed from the table for each ten revolutions, the point of the knife being entered different distances from the edge of the cylinder, the machine automatically feeding itself from a large bin above.

Depth of cut.	Number of kilos per 10 revolutions.	Depth of cut.	Number of kilos per 10 revolutions.
2 inches	133	10 inches	276
" "	133	" "	276.5
" "	135.5	" "	276
4 inches	178	12 inches	290
" "	176	" "	291
" "	175.5	" "	292.5
" "	175	" "	298
		" "	298
6 inches	218		
" "	218	14 inches	302
" "	218	" "	300
		" "	304
8 inches	258	" "	309
" "	258.5	" "	305
" "	258		

Cost of Making Concrete.

43. The cost of making concrete, where the materials were furnished near the job, at the prices mentioned below, and the concrete mixed by hand, has been found to average $5 per cubic yard.

No. 1 Concrete Experiment.—A Concrete Mixer was installed in a central position, where the stone, cement and sand could be elevated into hoppers by means of a system of buckets elevated on an elevator and pushed by hand at both the bottom and the top, the labor including shoveling the sand and stone into the buckets and emptying cement from the bags. The cost of this concrete was as follows:

1 cubic yard of concrete, with a 1—3—5 mixture, weighs 3,800 lb., and contains as follows—

4.5 cu. ft. or 1¼ bbls. of cement @ $1.15 per bbl.	= $1.54
13.5 " " of sand	= .45
22.5 " " " spalls, 1,975 lbs. @ $0.35 per net ton	= .345
For hauling sand, then elevating and placing in hopper	= .021
" " cement, then elevating and placing in hopper	= .025
Cost of mixing 50 yd. per day (labor) @ $.15 per hour	= .056
Current and motors	= .01
Repairs—Average for six months	= .01
Cost per cu. yd. loaded into wagons	= $2.457
Cost of making concrete exclusive of material	= .122

$2.457 ÷ 38 = $.065 per 100 lb.

Cubic feet of stone for 1 yard of concrete = 22.5 cu. ft.
" " " sand " 1 " " " .... 13.5 cu. ft.
To fill voids in stone 40 per cent ..... 9.0 "
4.5 "
Leaving excess sand ..... 4.5 "
Cement fills voids in sand ..... 27.0 "

44. The Concrete Machine running with the table making 10 revolutions per minute, has a capacity of 50 cubic yards per hour.

The above figures for labor are higher than they would be if more concrete had been made, only 50 cubic yards being the amount required.

The machine was operated by making two or three cubic yards at a time, and then stopping until more concrete was required.

No. 2 Concrete Experiment.—Foundations of a large building, the concrete being hauled in carts and dumped into trenches.

Cost of concrete loaded into wagons	= $2.45
Hauling from mixer to job	= .19
Putting in place	= .28
Total cost per cubic yard in place	= $2.92

In the above case the concrete was made as fast as six carts could haul it 500 feet.

No. 3 Concrete Experiment.—Cost of machine shop floor, having 8-inch thickness of concrete, left with a rough surface.

Cost of concrete loaded into wagons	=	$2.45
Hauling from mixer to job	=	.175
Putting in place	=	.86
Total cost per cubic yard in place	=	$3.485

No. 4 Concrete Experiment.—Heavy Foundations.

Concrete put into wheelbarrows and wheeled on plank runways about 150 feet.

Cost of concrete loaded into wheelbarrows	=	$2.45
Wheeling 150 feet and putting in place	=	.73
Total cost per cubic yard	=	$3.18

No. 5 Concrete Experiment.—Heavy concrete retaining walls, and concreting around a series of large pipes in a trench, hauling concrete 2,200 feet on bad roads, and wheeling from where wagons were dumped and putting in place.

Cost of concrete loaded into wagons	=	$2.45
Hauling 2,200 feet on bad roads	=	.26
Wheeling from wagons, dumping and putting in place	=	.22
Total cost per cubic yard	=	$2.93

45. The Superintendent of the plant in charge of this work reports that the Concrete Mixer has been in use about six months, and they find it is a great labor saver. The concrete is well made, both as to proportion and mixing, and careful checking of the amount of cement, sand and spalls used and fed to the machine shows that the machine is measuring the quantities accurately. It will be noted that the cost of making the concrete, exclusive of the materials, is only 12 cents per cubic yard. In other situations where the material is more easily handled this cost would be considerably reduced.

46. In conclusion it seems proper to state that the principles and details used in the measuring and mixing machines described have been made the subject of patents in this country and abroad.

No. 1091.*

SOME TYPES OF CENTRIFUGAL PUMPS.†

BY WM. O. WEBBER, BOSTON, MASS.
(Member of the Society.)

1. It is, perhaps, not generally known to the average reader that great improvements have been made in centrifugal pumping in the last decade, previous to which very little advance had been made for a long time. In fact the capabilities of centrifugal pumping seem to have been universally neglected. Perhaps a slight retrospect might not be out of connection in this place.

2. Centrifugal pumps were known as early as the seventeenth century. Papin, the celebrated French engineer, designed a centrifugal pump in 1703. Euler brought out a pump of this type in 1754. In 1818 a form of centrifugal pump was brought out in Massachusetts, U. S. A., known as the Massachusetts pump. In 1830 a Mr. McCarthy erected a pump in the New York navy yard which was credited to have approached the efficiencies of the present day. In 1846 Andrews produced pumps of this type, and about this same time John and Henry Gwynne of England commenced the manufacture of centrifugal pumps as a commercial enterprise. Appold exhibited a model of a centrifugal pump at Birmingham in 1849, and at the Crystal Palace Exposition in London in 1851, Appold's pumps were an important feature.

3. All of the pumps above referred to were pumps in which there was a single impeller or runner. Some of these pumps had

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* Presented at the Scranton meeting, June, 1905, of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussions on this general subject consult *Transactions* as follows:

No. 214, vol. 7, p. 598: "Relative Efficiency of Centrifugal and Reciprocating Pumps." Wm. O. Webber.

No. 274, vol. 9, p. 228: "Centrifugal Pumps and Their Efficiencies." Wm. O. Webber.

No. 763, vol. 19, p. 255: "Tests of Centrifugal Pumps." R. C. Carpenter.

No. 877, vol. 22, p. 262: "Tests of Centrifugal Pumps." W. B. Gregory.

the suction upon one side only, and some, like the Massachusetts and Gwynne pumps, had double suction inlets or openings on either side of the pumps. All of these pumps were made for comparatively low lifts and large quantities of water, and were not productive of very high efficiencies.

4. From 46 per cent. to 60 per cent. seems to have been the best results obtained with pumps of this type under heads varying from $4\frac{1}{2}$ to 15 feet. These pumps were directly connected to

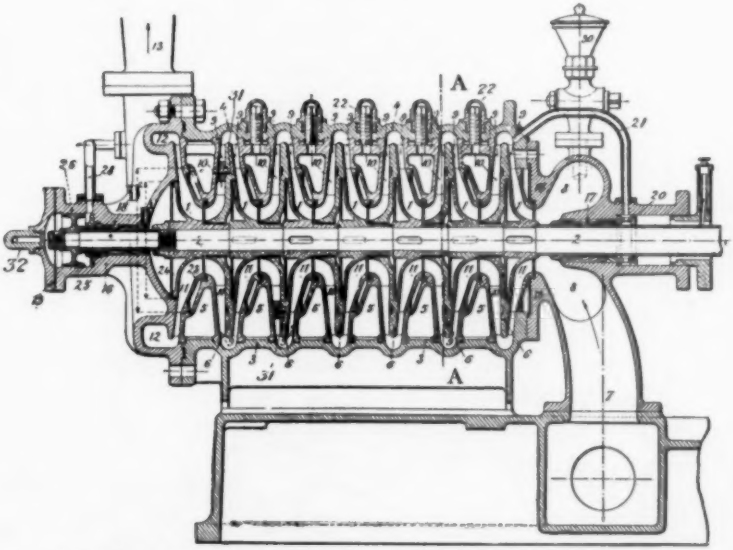


FIG. 266.—DETAIL OF THE A. C. E. RATEAU CENTRIFUGAL PUMP.

steam engines, and duties of from five and one-half million to forty-eight million foot pounds per one hundred pounds of fuel were obtained.

5. Forty feet was formally considered the maximum height at which centrifugal pumps could operate efficiently, and the experiments of the writer, conducted in 1874, 1875 and 1876, seemed to demonstrate that the highest efficiency in the single stage pump was found at a height of about 32 feet, and a maximum velocity of the liquid being pumped through the discharge orifice of the pump of not exceeding 12 feet per second.

6. In Appold's experiments he determined that the efficiency mainly depended upon the form of the blades in the impeller or

vane and the shape of the volute or enveloping case, and that the best form for the blade was a curve, pointing in the opposite direction to that in which the impeller revolved, and for the case, that of a spiral tapering type or volute.

7. The first engineer who seems to have discovered the value of compounding centrifugal pumps—that is, by causing the discharge of one centrifugal pump (either integrally formed or simply in close juxtaposition) to be led into the suction of another

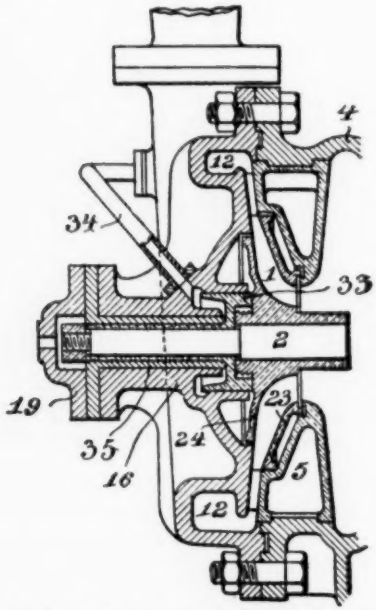


FIG. 267.—PARTIAL DETAIL OF THE A. C. E. RATEAU CENTRIFUGAL PUMP.

similar pump—was the celebrated Swiss engineer, Sulzer, of Winterther, Switzerland.

8. Sulzer was pretty closely followed by A. C. E. Rateau of Paris, France, Figs. 266 and 267, John Richards, Fig. 268, and Byron Jackson of San Francisco, California, Fig. 269. All of these pumps were originally constructed with the impellers facing in one direction, and the consequent thrust upon the impellers was partially taken up by balancing chambers upon the rear of each impeller. This form of balancing was further improved in Rateau's pump by a balancing cylinder at the extreme discharge end of the pump shaft, into which water was admitted from the dis-

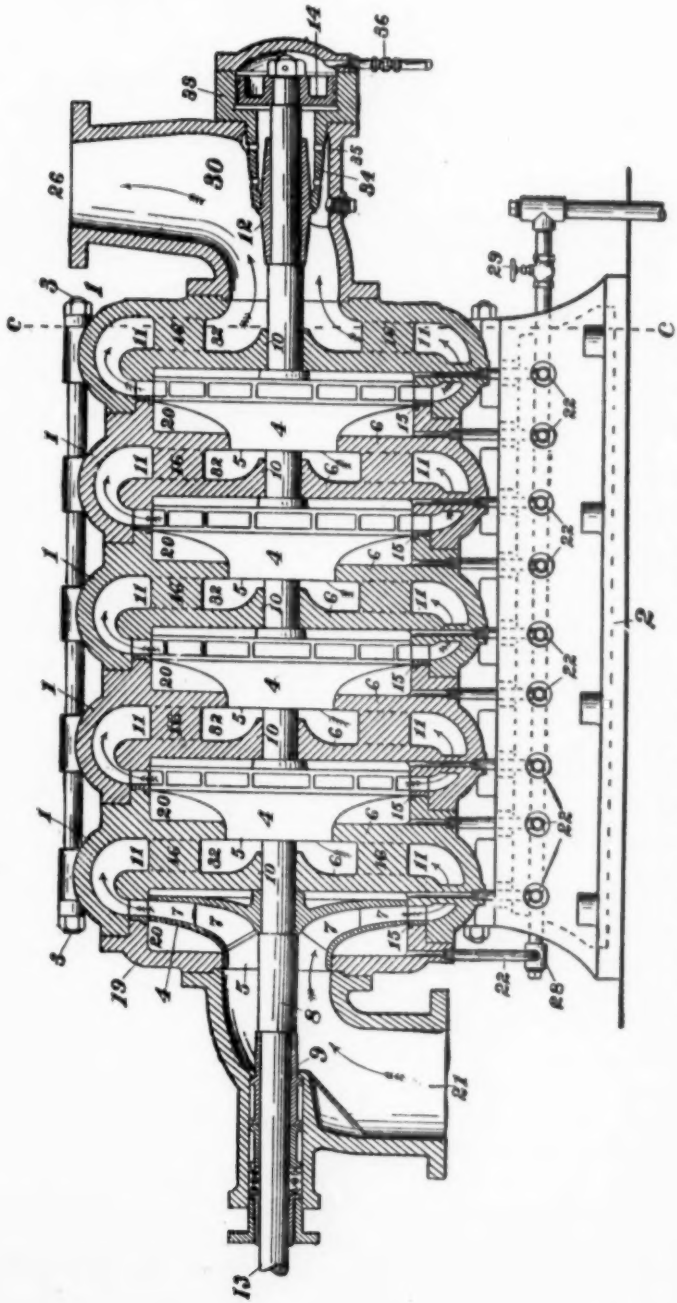


FIG. 268.—DETAIL OF THE J. RICHARDS HIGH-PRESSURE ROTARY PUMP.

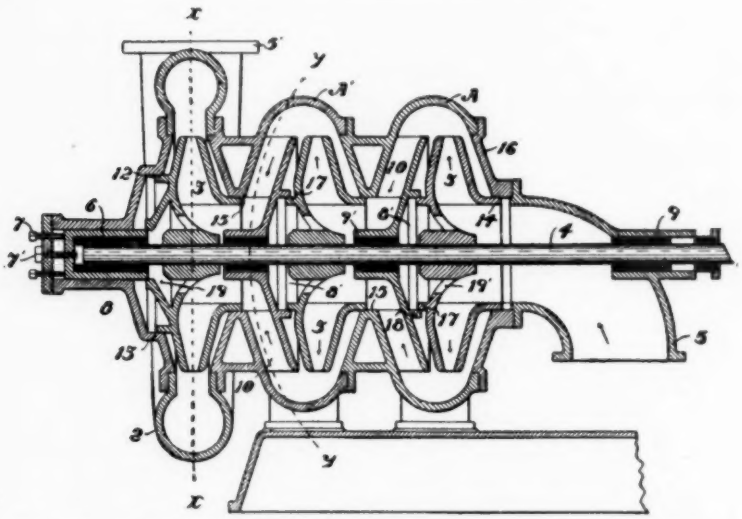


FIG. 269.—B. JACKSON'S SERIES HIGH-PRESSURE CENTRIFUGAL PUMP FOR HIGH HEADS.

charge orifice of the pump. Rateau further attempted to improve the balancing of his pump by constructing inclosed impellers, having one of the side plates of the impellers of smaller diameter than the other, Fig. 270, so that the area of the two impeller side plates subjected to the discharge pressure from that impeller was equal.

9. John Richards also used a balancing piston at the discharge end of the pump, directly connected with the discharge pressure, and also attempted to further balance a pump of this type by exhausting the air from one side of the impeller and forcing air in upon the other side. See pipes 22-28, Fig. 268.

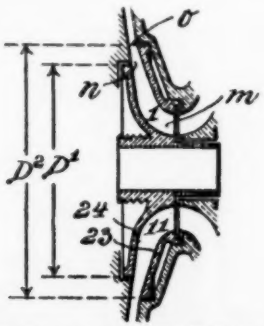


FIG. 270.—PARTIAL DETAIL OF THE A. C. E. RATEAU CENTRIFUGAL PUMP.

10. Another method of balancing the end thrust of centrifugal pumps was by false vanes located upon the outside of the side plates of an inclosed impeller, these being designed to relieve the pressure upon the outside of such impellers, by establishing a counter-flow through the space around the periphery of the impeller and causing a partial vacuum to exist. By making these false vanes of different radial lengths on the opposite sides of the impeller, the end thrust can be counteracted for any given set of conditions, but it is not believed that a pump thus provided with balancing vanes will remain in good operative balance if the original conditions are changed.

11. The diffusion vanes, which form a distinguishing feature of the turbine type of pump, are located in the usual whirlpool chamber of the centrifugal pump in such a manner as to convert

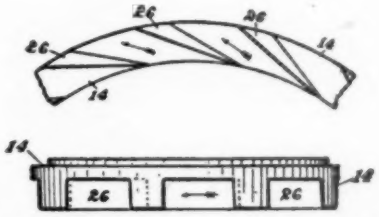


FIG. 271.—DETAILS OF THE J. RICHARDS COMPOUND ROTARY PUMP.

the water leaving the impeller at a high tangential velocity into a direct radial flow. This also has the effect of overcoming the eddies formed by the water leaving the impeller, thereby reducing the excessive internal friction.

This construction, therefore, causes a centrifugal pump to assume the converse of a turbine water wheel. The direct radial flow becomes of great importance in compound centrifugal pumps where the liquid being pumped is to be passed from one impeller to another. Many different forms of these diffusion vanes have been devised, and John Richards seems to have been the first one to form these diffusion vanes in a removable annular ring surrounding the impeller, Figs. 271 and 272, and as these parts are subject to considerable wear, their removal and replacement are thus rendered convenient and inexpensive.

12. Sulzer then made the greatest step in the balancing of compound pumps by placing the impellers back to back in pairs, so that one impeller thrust in one direction and the next impeller

thrust in the opposite direction. The method which he used, however, of conveying the liquid being pumped from one impeller to the next required short, tortuous passages, he utilizing the fixed guides in the whirlpool space of the pump to form these cross-over passages; also, it is believed, that to Sulzer we are indebted for the idea of introducing the guide blade or turbine element.

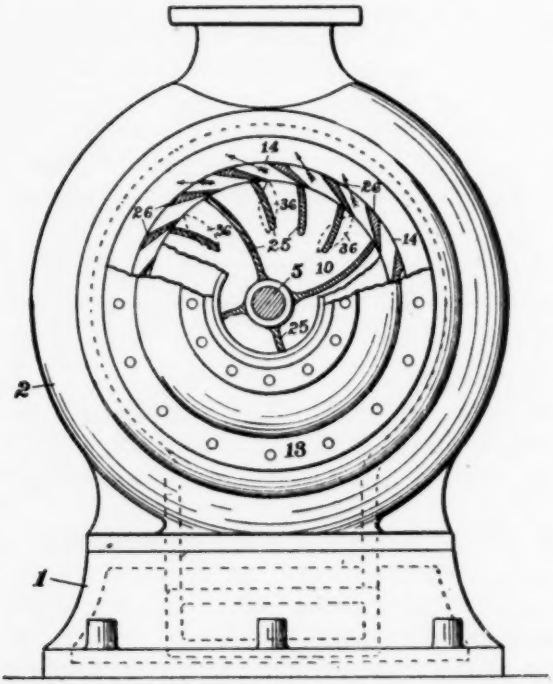
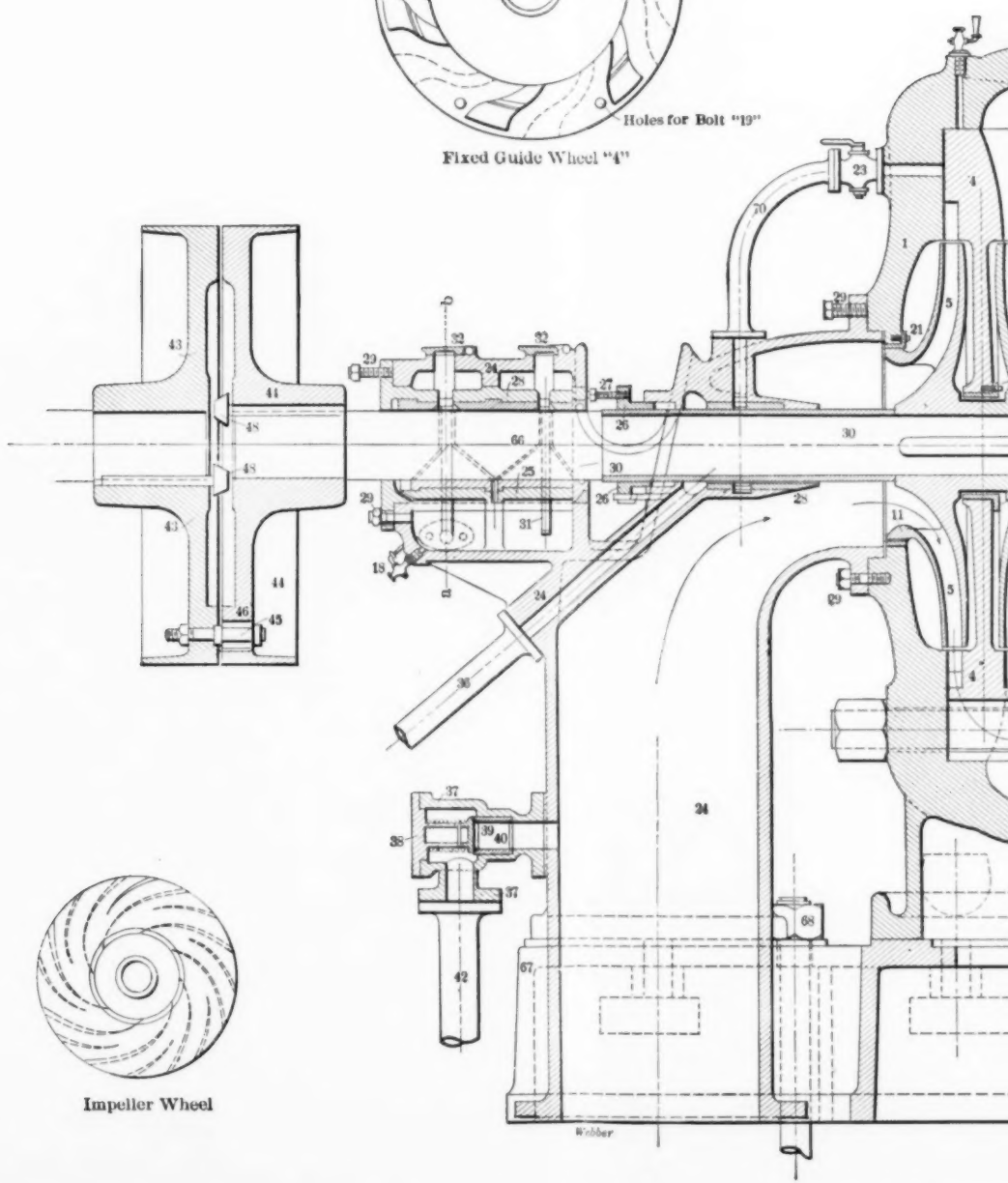


FIG. 272.—THE J. RICHARDS COMPOUND ROTARY PUMP.

13. John Richards followed Sulzer in placing two impellers back to back and differed from Sulzer in passing the discharge from one impeller out and around the discharge chamber of the final impeller, but only showed this construction in a two-stage pump. Fig. 274.

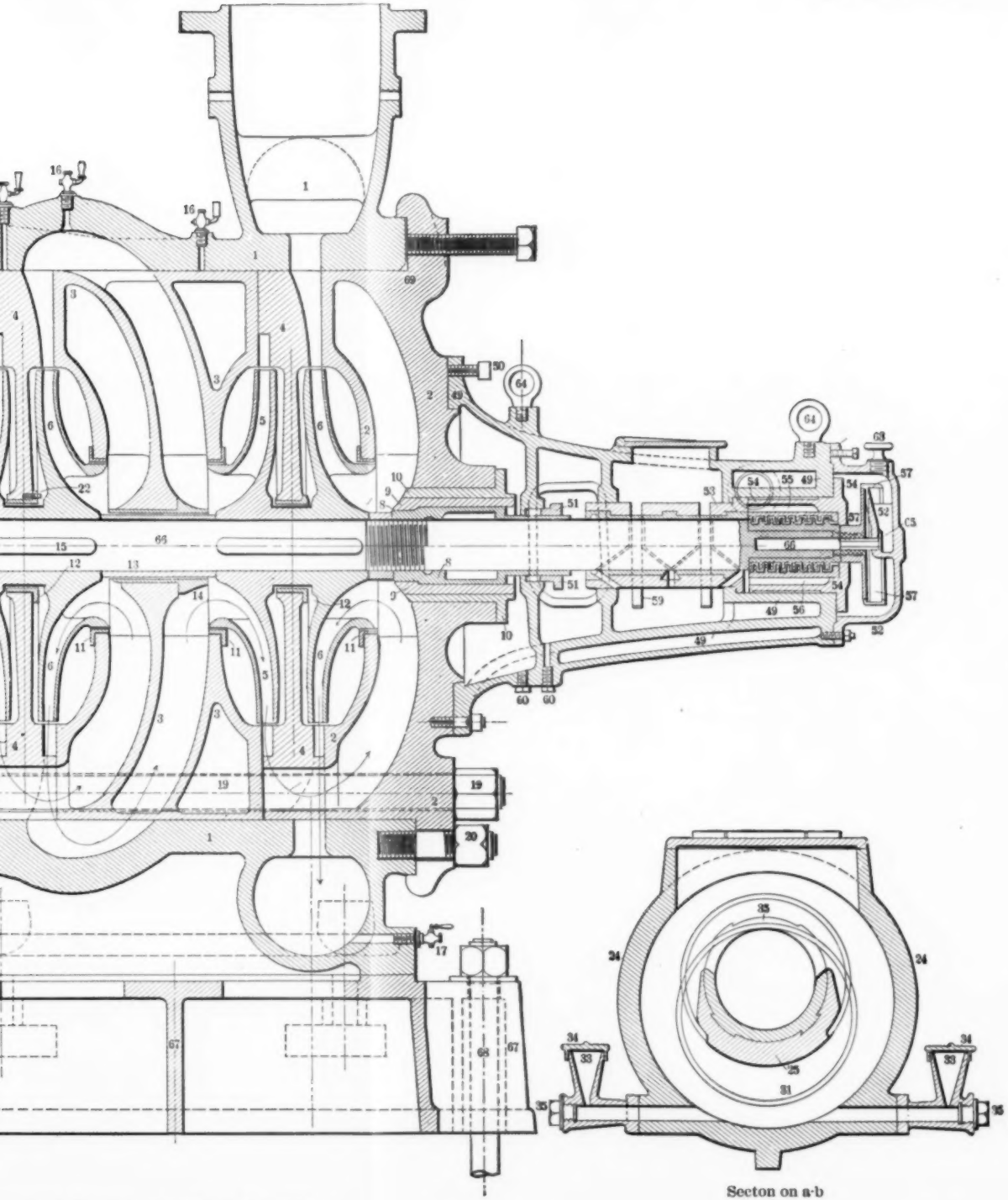
14. Hochdruck and Sulzer both locate their impellers back to back in pairs and pass the liquid from one impeller through lateral passages located between the radial vanes of the stationary guides surrounding the peripheries of each pair of impellers. In this form of construction, while the sum of the discharge pres-

Fixed Guide Wheel "4"



Impeller Wheel

FIG. 273.—HOCHDRU



Secton on a-b

HDRUCK CENTRIFUGAL PUMP No. VII.

sures in one direction balance the sum of the discharge pressures in the opposite direction any variation in the suction pressures will throw such a pump out of balance.

15. This is evidenced in that all of the constructions of Sulzer, Richards, Rateau and others show a thrust-bearing and balancing piston at the discharge end of their pumps, which they have speci-

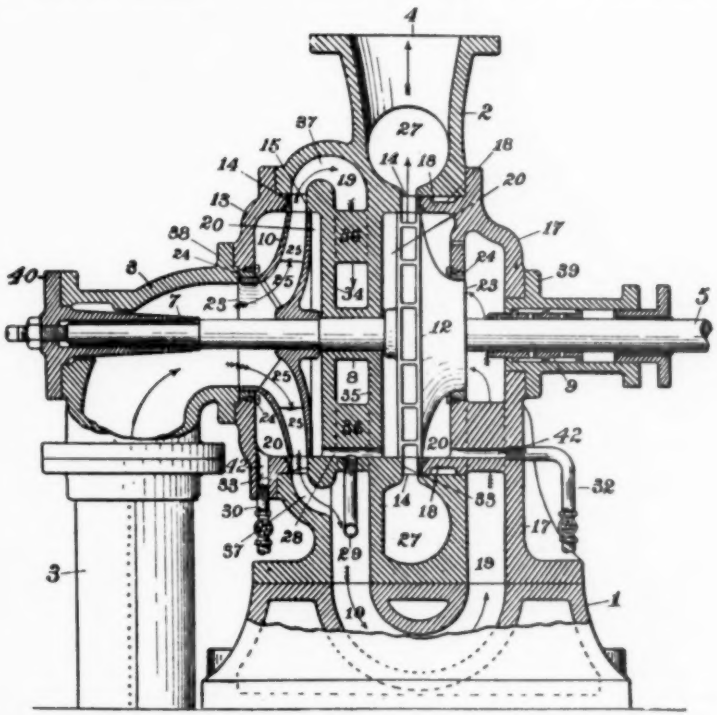


FIG. 274.—DETAILS OF THE J. RICHARDS COMPOUND ROTARY PUMP.

fically stated was put there for the purpose of overcoming the residual thrust.

16. It is the opinion of the writer that there should be no thrust, residual or otherwise, left over or unbalanced, in a properly constructed pump.

17. The writer believes that he is the first one to design an absolutely balanced compound centrifugal pump, by causing the impellers to be placed back to back, but not in pairs serially deposed; and in producing results, from the method in which he sequentially connects the different impellers, so that both the sum

and difference of the suction and discharge pressures of all the impellers in any compound centrifugal pump equalize and balance each other.

18. Some of these different types of pumps may be of interest.

Fig. 275 shows a half section of one of the writer's six-stage compound pumps.

Fig. 276 shows two spherical compound pumps coupled in series for very high lifts, the motor in the center, driving both pumps

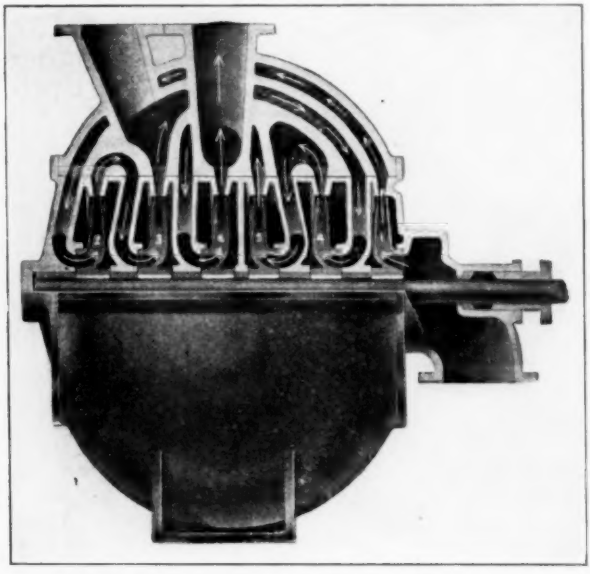


FIG. 275.—10-INCH SIX STAGE SPHERICAL PUMP.

at a constant speed, the second pump of the series being made practically twice as heavy as the first pump of the series, which takes the water practically under no pressure.

Fig. 277 shows a four-stage spherical pump coupled to a 50 kw. motor.

Fig. 278 shows a very large two-stage spherical pump, directly connected to a 1,400 horse-power alternating current generator, used as a motor.

Fig. 279 shows a ten-inch, ten-stage spherical pump, directly connected to a 10-20 by 12 vertical cross-compound engine.

Fig. 280 represents a vertical turbine sinking pump, directly

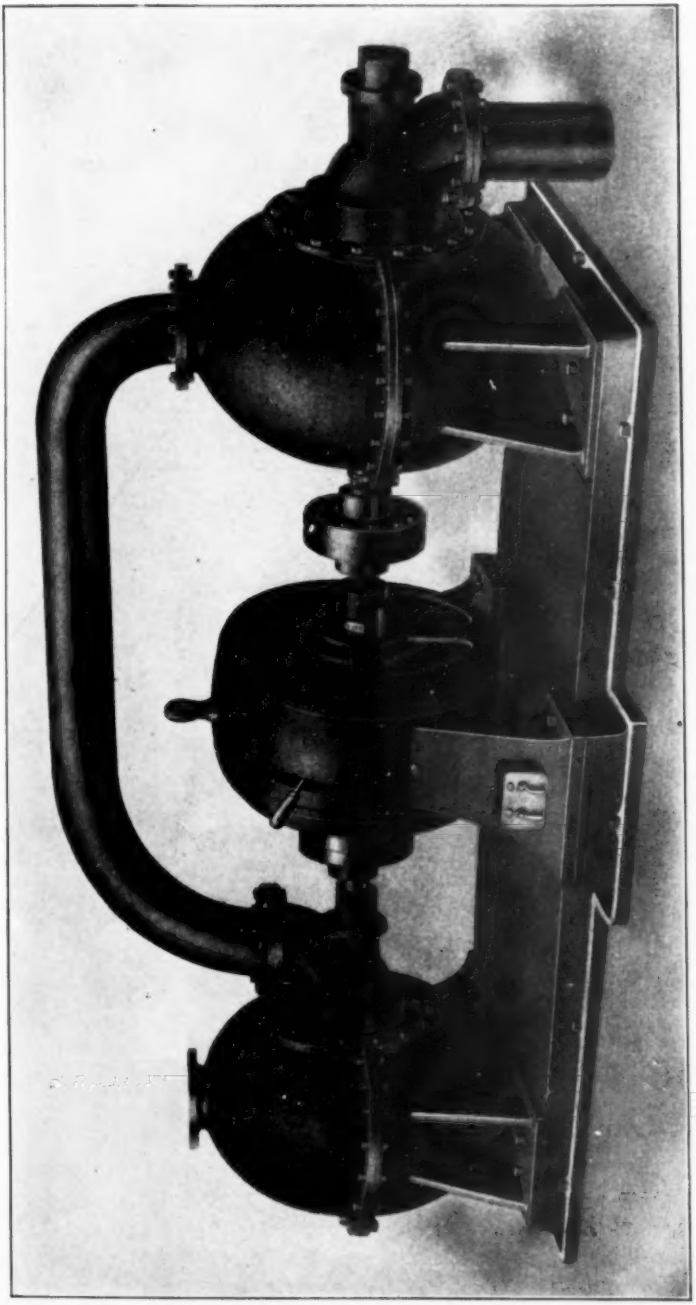


FIG. 276.—TWO SPHERICAL COMPOUND PUMPS IN SERIES FOR HIGH LIFT WITH MOTOR IN CENTER.

connected to a vertical motor, enclosed in a cast iron housing. This form of pump is designed to be lowered into a shaft and to follow the work of excavation as the shaft is carried down.

Fig. 281 represents a vertical turbine pump in the bottom of a shaft, directly connected by a vertical shaft to a horizontal tandem compound steam engine with vertical pillow-block bearing.

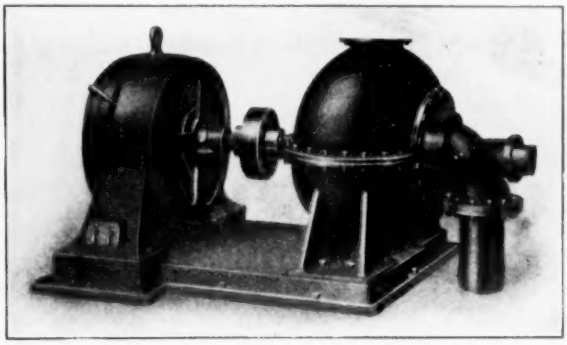


FIG. 277.

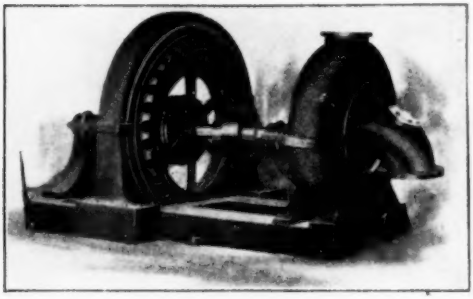


FIG. 278.

19. Carl Lager, of Syracuse, New York, has also designed a pump in which the sum and difference of the suction and discharge pressures of all the impellers in any compound centrifugal pump equalize and balance each other, but accomplishes his result by placing his impellers in pairs, facing each other, with a diaphragm on an angle between the two impellers, he confining his discharge from one impeller to a certain locality upon the periphery of the pump and not entirely around the periphery.

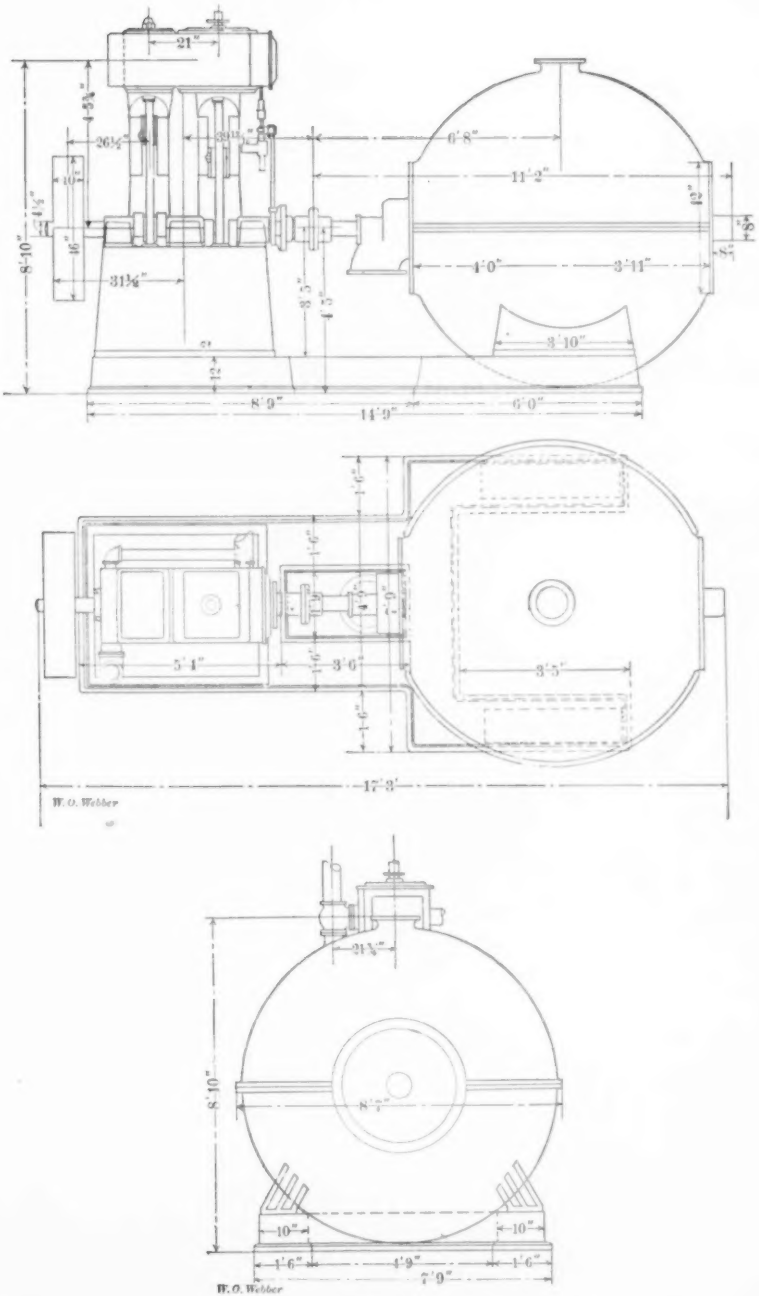

21"
26½"
10"
16"
31½"
6'8"
11'2"
4'0"
3'11"
8'10"
8'9"
14'9"
6'0"
5'4"
3'6"
8'5"
17'3"
W. O. Webber
21½"
8'10"
10"
1'6"
4'9"
7'9"
W. O. Webber

FIG. 279.

20. Still another interesting feature in regard to compound centrifugal pumps is the requirement to discharge a given quantity of water to a desirable height, and then twice the quantity of water to practically one-half of the original height without changing the speed of the pump.

21. This requirement is met with in barometric condensers where it is required to discharge sufficient water over the top of

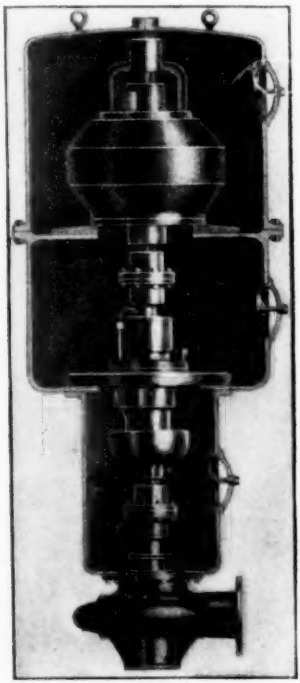


FIG. 280.

an inverted syphon to start the syphon, and then, having started the syphon and receiving the benefit of its action, to supply at least double the amount of water to the lower head required when the syphon is in action.

22. This is accomplished by placing two centrifugal pump impellers back to back, providing each with its own suction and discharge passages, but also providing a passage by which one impeller may discharge into the suction of the second impeller, and further providing valves by which the independent discharge of

the first impeller may be cut off and *it* forced to discharge through the second impeller, and the supply of the second impeller cut off and *it* forced to take the discharge of the first impeller, by which

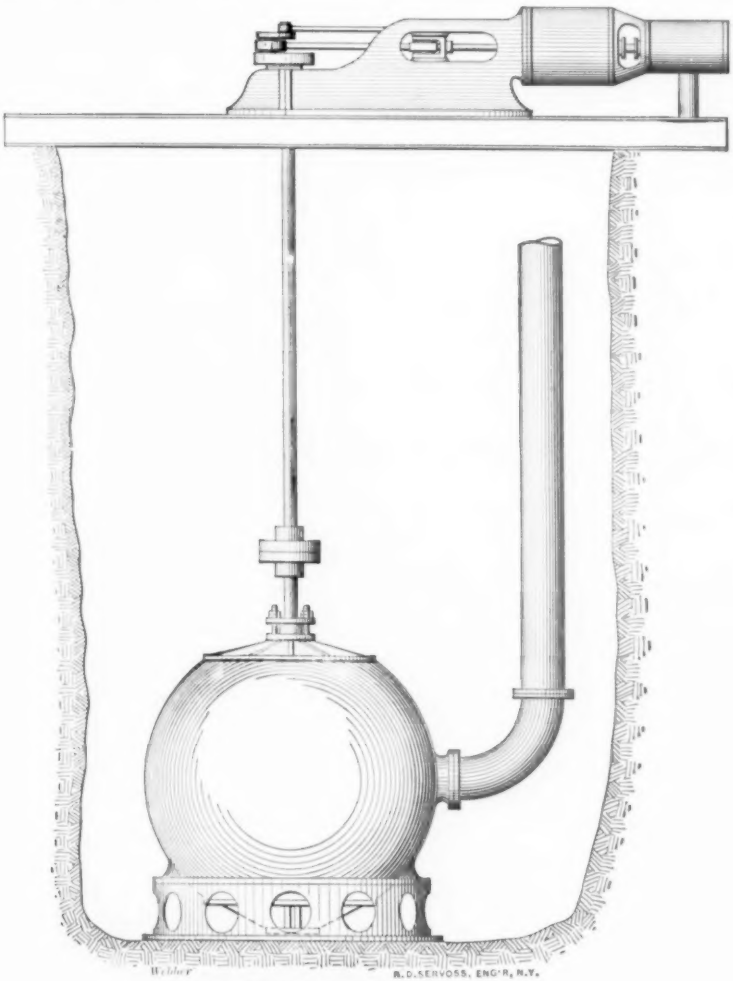


FIG. 281.—VERTICAL TURBINE PUMP.

arrangement the two impellers may be run in parallel or series, as may be desired.

23. Still other improvements have been made in which multiple stage pumps are provided with a plurality of inlets, or outlets, as

the case may be, so that the pump at will may be made to discharge from any of its impellers. This system was devised to have a fire pressure and a tank pressure supplied from the same pump at will, or to take care of an ordinary flow at a point of maximum economy and have the same pump take care of a storm overflow for a temporary period, without relation to economy—this being done to avoid the unnecessary expense of installing a second complete unit.

24. With the present type of multiple stage centrifugal pumps this pump at once offers the solution of pumping large quantities of foul, or dirty water, often impregnated with foreign material, to a considerable height, such as the pumping out of iron, coal and other mines.

25. The combination of this type of pump, directly connected to high-grade compound condensing steam engines, provided with automatic cut-offs, presents an economical type of waterworks pumping engine, occupying approximately one-sixth of the room of the ordinary high duty pumping engine, and costing one-sixth the price, and giving duties very nearly in comparison to the best engines of these types.

26. For instance. A 20-inch six-stage pump directly connected to a 15-30 by 16 compound engine, with steam at 200 pounds, would easily give a duty of 90,000,000 foot pounds per 1,000 pounds of dry steam. This pump would be capable of lifting 20,000,000 gallons of water, per day of twenty-four hours, to a height of 185 feet, and with 100 degrees of superheat and 28 inches of vacuum, such an outfit would readily give from one hundred to one hundred and five million foot pounds duty.

27. These pumps, directly connected to electrical motors, present practical solutions of many unique problems where it is desired to pump water to extreme elevations.

The writer recently figured on a plan of this sort, to pump a thousand gallons of water a minute to a height of 6,000 feet, using five compound units, each pumping 1,200 feet, two pumps working in series constituting each unit with a motor between, as shown in cut herewith. This insulation was designed for furnishing a water supply for the British summer capitol at Simla, India.

Two questions are continually coming up in relation to centrifugal pumps:

28. One is the revolutions per minute required to discharge any given quantity of water to a given height, and the second is the

necessary reduction of speed required to discharge a smaller amount and the necessary increase in speed required to discharge a larger amount than the normal, or most efficient capacity of a pump.

29. The old formula of Appold, which was

$$\sqrt{\frac{Hf \times 550 + 500}{Cf}} = \text{R. P. M.}$$

in which Hf was the static head in feet, and Cf the circumference of the impeller in feet; the quantity 500, used as a multiplier, was supposed to be the $\sqrt{2gh} \times 60$ equal to 480 feet per minute plus a constant necessary to overcome internal friction, and the constant 550 the additional velocity in feet per minute necessary to give some indefinite quantity of discharge.

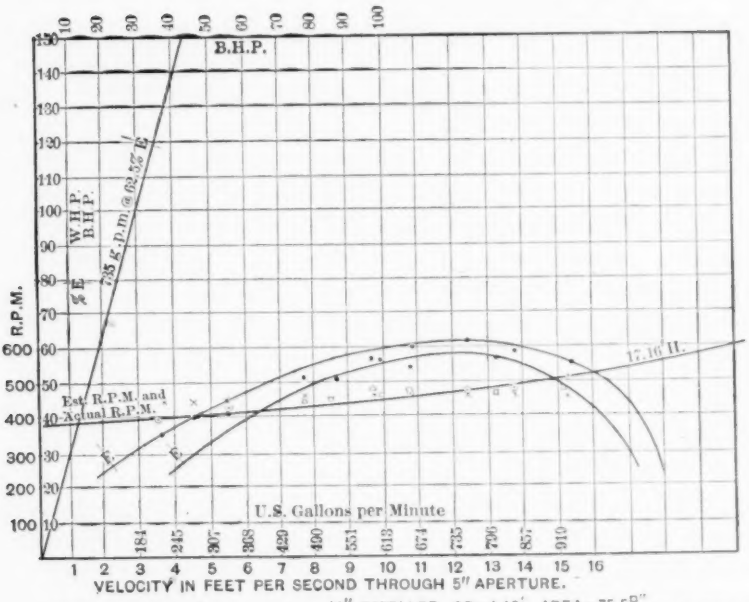
30. All of the writer's experiments have shown that, probably owing to the water in the whirlpool space actually pumping water, which has the effect of increasing the actual diameter of the impeller to a diameter equal to about half the depth of the whirlpool space, the square root of $2gh$., for pumps with whirlpool spaces, should be equal to about 425.

31. It was also further demonstrated that the width of the opening at the periphery of the impeller had a great deal to do with determining the amount of the constant to be added for any required amount of discharge, and the writer evolved it into an equation like this:

$$\frac{D \times 3.1416 \times W}{\varphi} = \text{r} \qquad \frac{V \times 60}{r} = \text{c}$$

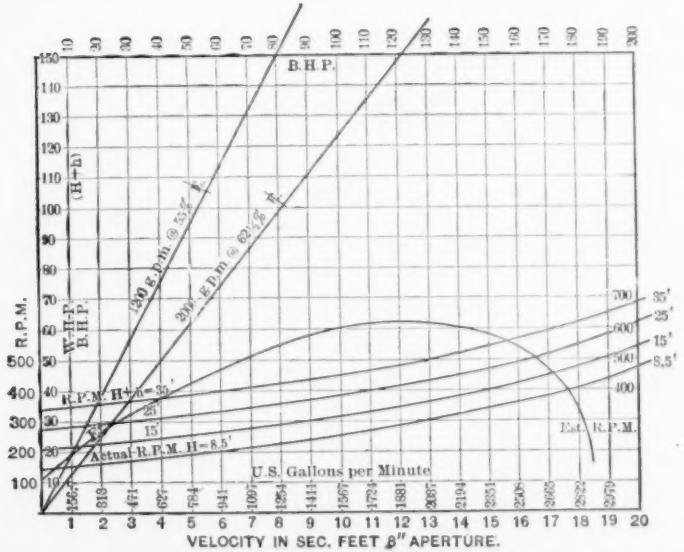
in which D equals impeller diameter; W equals width of opening at periphery of impeller, and ϕ equals area at the neck of the discharge aperture of the pump; the resultant r being the ratio that these two bear to one another, and in the second formula V equaling the velocity in feet per second through this same discharge aperture; the resultant c being the actual feet per minute of peripherical speed necessary for the discharge represented by V.

32. These formulas give a speed curve which would be correct if the resistance due to the velocity of discharge was constant, which, however, is not a fact, so that empirical percentages must be added to, or subtracted from, this result, as the discharge is above or below the normal speed.



TESTS ON 5" B. PUMP. 16" IMPELLER CF=4.19. AREA=75.5$^{\square''}$ 3.8 RATIO.
5" AREA=138$^{\square''}$=19.80$^{\square''}$
Webber. Wm. O.

FIG. 282.



TESTS OF 8" B. PUMP. 24" IMPELLER=6.28. AREA=113.
8" AREA = .347$^{\square'}$ = 50.25$^{\square''}$=2.26 RATIO
Webber. Wm. O.

FIG. 283.

33. Fig. 282. The line marked "estimated R. P. M." represents the theoretical, and the line marked "actual R. P. M." the revolutions per minute required to discharge the quantities of water through a 5-inch aperture, at an elevation of 17.16 feet. In other words, the "estimated" line would represent the revolutions per minute for static head H, whereas the "actual" would represent the R. P. M. at the dynamic head, or $H + h$.

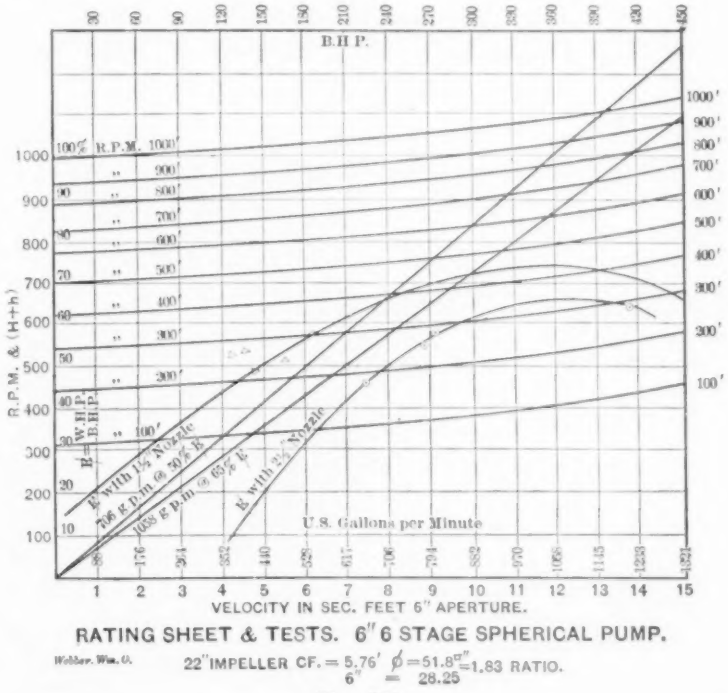

B.H.P.
R.P.M. & (H+h)
U.S. Gallons per Minute
VELOCITY IN SEC. FEET 6" APERTURE.
E with 1¾" Nozzle
E with 2¼" Nozzle
RATING SHEET & TESTS. 6" 6 STAGE SPHERICAL PUMP.
Webber, Wm. O.
22" IMPELLER CF. = 5.76' ϕ = 51.8"/28.25 = 1.83 RATIO.

FIG. 284.

34. These figures are based on the test which was the subject of a previous article published by the writer in vol. 7 of the *Transactions*, in which the ratio r between the area through the periphery of the impeller and the discharge area of the pump was as 3.8 is to 1.

35. Figs. 283 and 285 represent actual revolution curves for given discharges for larger apertures, and show the relation between such revolutions at different heights of discharge.

36. Fig. 284 is a rating sheet made from the tests of a 6-inch six-stage spherical pump in which the ratio r is 1.83 to 1, and gives

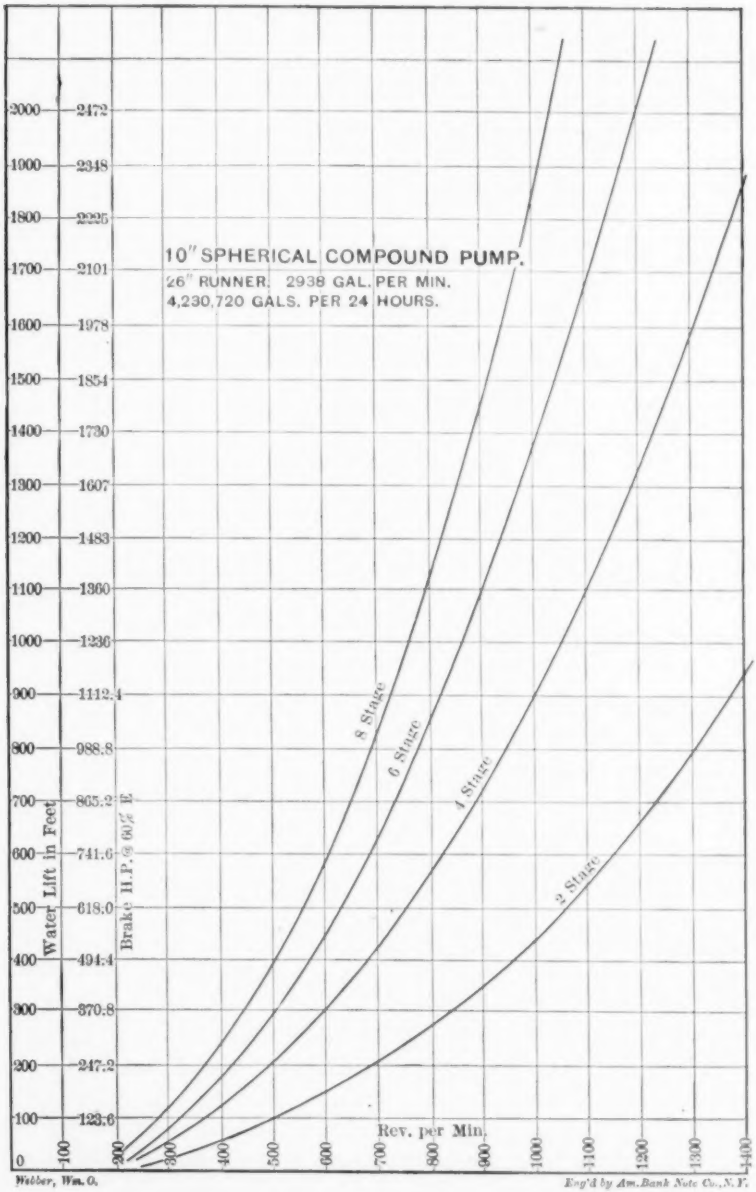


FIG. 285.—HORSE-POWER AND SPEED CURVES COMPOUND PUMP.

these speed curves up to heights of 1,000 feet. This diagram also shows the efficiency curves obtained by throttling the discharge by means of Fleigner nozzles, the discharge, however, in these tests being taken from the weir reading and not depending on the pressure due to the nozzle, excepting that this pressure was taken to represent $H + h$.

TESTS OF 6 INCH 6-STAGE SPHERICAL PUMP.

Rev. of pump		365	370	400	440	470	430	635	800	810	816
Dynamic height		102	104.3	166.2	182.5	194	205.3	342	540	563	587
U. S. gallons per min.	760	772	780	560	590	610	628	680	750	800	850
Water horse-power	19.05	19.85	21.6	23.55	27.2	29.97	32.5	58.5	101.5	113	124
Brake horse-power		42.5	42	35.6	40	47.2	54.8	105	173	199	202
Efficiency $\frac{\text{W.H.P.}}{\text{B.H.P.}}$	45	46.8	49	66	68.2	63.5	59.4	56.5	58.7	56.7	61.5
Pressure on pump		40	41	68	75	86	85	140	225	235	245
Diam. of dis. nozzle		2¼	2¼	1½	1½	1½	1½	1½	1½	1½	1½
Pressure at nozzle		40	41	68	75	80	85	104	115	136	160
Suction, dynamic		10′	10′	10′	10′	10′	10′	19′ 2″	19′ 2″	19′ 2″	19′ 2″

37. It is interesting to note how the efficiencies increase at the smaller quantities when pumping at the higher heads.

38. In this connection would state that we obtained an efficiency of 82½ per cent. at 12 feet velocity in a single instance, but, not being able to repeat the experiment, the writer is still doubtful as to its accuracy.

39. It must be understood, of course, that in all experiments of this nature the form of the curve of the impeller blade plays a very important part. The writer, by his own experiments, demonstrated that a curve of rolling friction gave a much better result than any of the curves described and advocated by Rankine, Appold or Unwin, and his data are based upon the curves evolved from his own experiments. He is well aware that other results will be obtained from other curves, and has devised other forms of curves to be used with extremely high revolutions per minute, in which entirely different formulas than those quoted have to be used.

40. He finds that these other curves, although necessary for certain practical purposes, are always used at a sacrifice of efficiency. He is also aware that these formulas which he has referred to are necessarily crude, involving, as they do, the empirical additions and subtractions for velocities above and below a certain point, and advances the results he has in the hopes that some disciple of Euclid can evolve a more rational form of formula. His only other excuse for publishing them is that they work in practice

and check exactly results obtained by officers of the Navy Department, and other observers beside himself, on entirely different series of tests on similar pumps.

41. The multiple-stage centrifugal pump therefore offers a practical solution for pumping impure liquids to great heights or against great pressures, easily and economically, as in mines for pumping water containing all kinds of foreign materials for fire pumps to maintain a sufficient number of fire streams indefinitely.

42. Also certain combinations of multiple-stage pumps can be readily made to pump a quantity of water to a certain height to start a barometric condenser and then twice the quantity to one-half the original height after the siphon action of condenser is established to maintain the necessary vacuum at a constant speed of pump.

43. Other adaptations allow of maintaining a variation of pressures on hydraulic elevators at a constant speed of pump to vary the speed of elevators to accommodate variations in the demands for service during the different hours of a day. Or discharging under several different pressures for varying services or to varying heights for similar purposes. In fact the variations to be obtained by this class of pump seem to be indefinite and only limited by the demands for service and the ingenuity of the designer in making combinations of the different stages, consequent upon a multiplicity of impellers.

44. This type of pump has come to stay and offers a solution for sinking pumps in deep workings, for submerged pumping, for auxiliary pumps for water works, and many other combinations where electrical driving will allow of operating at a distance and without local attendance. They are also, from their extreme simplicity, absence of valves, etc., as nearly impossible of getting out of order and as nearly "fool-proof" as it is possible to make modern machinery. They occupy a minimum amount of space, can be operated in any position, can be made infinitely strong, do not get out of repair readily and can be repaired easily and the working parts renewed readily at a minimum expense.

DISCUSSION.

Mr. F. V. Henshaw.—Mr. Webber's paper is a timely contribution, as the possibilities of centrifugal pumps are beginning to receive general recognition. The new types designed on the

multiple stage principle extend the working range of this class of machinery to such an extent that we are no longer confined to reciprocating pumps for moderate and high heads.

Possibly the most important result of this development is that of greatly increased facility in the use of electric power for pumping.

Outside the boiler room itself electric power is fast becoming universal, and anything which adds to the adaptability to electric power of any operation is of value for that reason alone.

In driving reciprocating machines, the valuable characteristic of uniform rotary torque of the electric motor becomes naturally a positive disadvantage. Certain speed limits are imposed on electric motors by reason of cost and efficiency. These limits are much higher than the corresponding speeds of reciprocating pumps, hence double reduction gearing is often required. Pumps with less than three cylinders are not suitable for electric drive, and even with that number the jar is objectionable.

The centrifugal pump, on the contrary, lends itself perfectly to the speeds and characteristics of the electric motor, and by variation in the number of stages it would seem possible to meet most if not all conditions of speed and head with economy and convenience.

Available data on the efficiency of multiple stage centrifugal pumps are at present incomplete, and I hope that Mr. Webber and other workers in this field will soon favor the society with additional information on this point.

Enough is known, however, to indicate that with its other advantages the centrifugal or turbine pump stands a fair chance of entirely superseding the reciprocating pump for general work in lifting water.

It has even a possibility of competing with the most efficient steam machine known, viz.: the high duty waterworks pumping engine, by reason of its low cost and the indirect advantages due to its adaptability to electric power.

Consider a water supply system having a number of pumping stations in which are electric centrifugal pumps driven from a central station through high tension power lines. Against the reduced "duty" in such a system there would be all the advantages of concentrated steam plant at the most desirable location, steam turbines, ability to combine in the same station power

supply for municipal or general lighting and traction, and small cheap pumping stations which could be made nearly automatic.

There are certainly conditions under which such a system should receive serious consideration. In this age of "mergers," electric power in the engineering field is like the powers of Wall Street in another field; it tends to merge things, but unlike the latter, takes but a small percentage for its services.

In further reference to pumping systems, I would ask Mr. Webber if centrifugal pumps have been used or considered for oil pipe lines? Such pumps would seem well adapted to the work unless the pressures and the nature of the fluid prevent. For example, it would be interesting to know whether the agitation of the fluid by the impellers would prevent the deposit of wax in the passages of the pump.

An obvious advantage of such pumps for this work is the ability to isolate the pump from its driving power simply and efficiently, thus reducing fire risk.

Mr. Charles N. Trump.—Mr. Webber gives the names of a number of engineers who have worked at centrifugal pump building, but I find no mention of a plant designed by a member of this Society, the late John H. Cooper.

During May, 1898, I was presented with a manuscript* endorsed, "The Centrifugal Pumping Plant, at Mare Island Navy Yard, California, by John H. Cooper, M. E.," and for record in the Proceedings I quote as follows:

"A marked departure in centrifugal pump building in this country was inaugurated in the year 1884.

"During this year an example of superb stone dock architecture was nearing completion within the extended precincts of Mare Island Navy Yard, California.

"For emptying this dock of its seventy-nine millions of gallons of water, within the practical limits of time necessary to safely secure a vessel therein, there would be required an adequately proportioned pumping 'mechanism.'

"To this end, proposals were issued by the United States Government for a plant, to consist of a pair of centrifugal pumps having an aggregate average discharge of 80,000 gallons per minute, and a drainage pump with an average delivery of 2,000 gal-

* Paper read April 14, 1893.—Printed in *Journal Franklin Institute,* Vol. cxxxviii, pages 251 to 258.

lons per minute, together with the necessary pipes, valves, etc., to connect them with the suction wells of the dock and to the discharge culvert . . . including propelling engines, bed-plates supporting them, and full 'equipment for operating the same.'

"No general plans or details were furnished; the plant was to be delivered and erected in a house to be prepared by the Government, beside the dock, to which steam would be conveyed from boilers furnished by the department.

"This contract, awarded to the Southwark Foundry and Machine Co., of Philadelphia, bears date July 31, 1884, and to the writer of this article (John H. Cooper) was entrusted the entire formulation of the design and the conception and arrangement of its details. . . ."

Two 42-inch centrifugal pumps—each pump directly connected with an engine having steam cylinder, 28-inch diameter, 24-inch stroke *—were installed, for emptying the great bulk of water from the dock.

One 8-inch centrifugal pump—its engine with steam cylinder, 12-inch diameter, 10-inch stroke—was put in for completing the discharge and for continuously removing water of infiltration from the dock.

The two main pumps were tested together by the board appointed for that purpose, April 20, 1886. As quoted by Mr. Cooper, their report reads:

"During a pumping period of fifty-five minutes the dock had been emptied from the twenty-third to two inches above the sixth altar, containing 6,210,698 gallons, an average throughout of 112,698 gallons per minute. At one time, when the revolutions were increased to 160 per minute, the discharge was 137,797 gallons per minute." †

"In practice it would be impossible to secure a vessel in the dock—so rapidly was the water removed—at the time of trial, and it is without parallel in the world that so large a dock, with its 7,000,000 gallons of water, has been emptied in fifty-five minutes. One pump will do all that is required to empty the dock in use, and then it will require sharp work to properly shore the vessel, to replace the rapidly receding water. . . .

* Engines and pumps mounted on attached bed plates, forming one firmly united and self-contained whole.

† Head varied from nothing at start to 35 feet at finish.

"Indicator cards taken at various intervals gave 796 horse-power for the main engines. The revolutions did not exceed 160 at any time, though it was estimated that 900 horse-power and 210 revolutions would be necessary to attain the requisite delivery.*

Test of the engine-driving drainage pump developed 32 horse-power, steam pressure 55 pounds per square inch, 350 revolutions per minute. This pump gave records beyond requirements of contract, and did all its work under extreme head, 35 feet.

The plant was afterwards duplicated by the Southwark Foundry and Machine Co., at Pensacola, Florida.

Mr. Fred Ray.—There cannot be much doubt but that the credit of constructing the first compound pump belongs to some one of the various designers who have been developing this type of pump in California for the last twenty-five years or more. Just who, I am unable to state, but would call your attention to a patent granted to John Richards, March 31st, 1885, which shows a two-stage pump exactly similar in principal to the stage pump of to-day.

The author of the paper under discussion has gone into the subject of balancing quite extensively, but still has omitted much that is of interest. The methods of balancing shown in Figs. 266, 267 and 268 are almost the same, and are neither theoretically nor practically correct for the reason that the pressure which causes the unbalanced thrust is the difference between the suction and the periphery of the impellers, whereas the pressure water for the balancing piston is obtained from the discharge of the pump, and has been affected by the diffusion vanes of each stage. Consequently, in the case of Richard's pump shown in Fig. 268, the area of the balancing piston would be in the same ratio to the area of the suction as the total head generated at the periphery by any given impeller is to the total head generated between the suction of the given impeller and the discharge side of its diffusion vane. Consequently, for a pump of this design to remain balanced, it would require the ratio of these pressures to remain constant under the various capacities which the pump would be expected to deliver. This, however, is not the case, as can be readily determined by anyone who wishes to place gauges at these two points. This ratio is the greatest when the direction of the

* Plant tested, April, 1886.

flow of water from the impeller coincides with the angle of the diffusion vanes and becomes less as the quantity of water is increased or decreased from these conditions. Thus it is evident that such a pump could only be balanced under certain limited conditions. The same reasons apply to the balancing of the Rateau pump, only that it is further complicated by cutting down the diameter of the back side of the impeller as shown in Fig. 270. This construction also reduces the efficiency of the pump, owing to the additional friction loss which is caused by the impeller being partially of the open type.

The balance arrangement of the Byron Jackson pump shown in Fig. 269 is objectionable, owing to the fact that it requires end motion of the shaft in its operation, and consequently cannot be applied to the modern high efficiency turbine pump, as it is essential that the impeller openings shall coincide exactly with the opening in the different vanes.

There can be no question but that single suction impellers mounted in pairs with the suction openings reversed can be made to operate in perfect balance as far as end thrust is concerned. The thrust that each impeller exerts is caused by the difference of pressure generated by the impeller between its suction and periphery. Consequently, two impellers made identically the same, operated at the same speed and pumping the same quantity of water, will undoubtedly generate the same difference of pressure between their suctions and peripheries, provided the fluid which they are pumping is non-compressible. It thus seems reasonable that if these two impellers are mounted as shown in Fig. 274, the thrust of one will equalize the thrust of the other. Since it is the difference of pressure between the suction and periphery that causes the end thrust, the actual pressure on the suction of either impeller is immaterial. The reason that thrust bearings are usually provided with these pumps is due to the fact that it is practically impossible to construct the two impellers identically the same, and also due to the fact that the leakage of the water between the various fits in the pumps exerts a great influence on the pressures which cause the end thrust. It would be quite possible to take the pump shown in Fig. 274, and by slightly altering the fits around the impeller suctions, cause an end thrust of considerable magnitude to occur in either direction. It can be readily seen from this that no matter how perfect the pump may be constructed when new, the effect of wear

is liable to cause an end thrust after the pump has been in actual service. It is also evident from the above that it makes no material difference in what sequence the impellers are arranged in multi-stage pumps as long as half of the suction openings point in one direction and the other half in the opposite direction. If single suction impellers arranged in pairs will not operate in balance, as the author of the paper states, we would like to know the arrangement of the impellers in his two-stage spherical pump.

The method of balancing, consisting of placing a balance chamber of equal diameter to the suction opening on the rear of each impeller and having passages connecting to the suction, is the best method known for balancing the end thrust of the impellers, as is shown by the fact that it is used by some of the leading pump builders of both this country and Europe. The writer would be pleased to show anyone that is sufficiently interested in the matter to visit the works of Henry R. Worthington, Harrison, N. J., pumps operating in perfect balance having impellers either single suction mounted in pairs or single suctions with balance chambers on the rear. The spherical pump illustrated by the author of the paper seems to have been designed to obtain the arrangement of impellers which the author of the paper believes to be necessary to obtain a balanced end thrust. This design, however, seems to make an extremely large, heavy and expensive pump, and in the opinion of the writer accomplishes no results but what can be obtained by the far simpler construction in use by all of the best-known pump manufacturers.

One of the most important points in the design of high speed, high pressure turbine pumps is the shaft bearings. To operate satisfactorily, these bearings must be entirely removed from any possibility of contact with the water being pumped, and they also must be provided with self-oiling arrangements, and otherwise made equal to the bearings which are furnished with all classes of high speed electrical machinery. This seems to have been entirely neglected in the design of the spherical pump as shown, although such bearings have been furnished for sometime past by all the leading manufacturers of pumps all over the world.

The requirements met with Barometric condensers can be fulfilled by means of a single impeller pump if such is properly designed. It is standard with the Henry R. Worthington Company to furnish single impeller volute centrifugal pumps with

their Barometric condensers of such a design that they will deliver about half their normal capacity over the top of the syphon, and their full capacity when the vacuum in the condenser is obtained, the speed and power remaining constant. Pumps of this type can be seen in continuous operation at the Power Station of the United Electric Railways in Baltimore.

It might be well to call your attention to the fact that multistage pumps provided with a plurality of inlets are covered by U. S. patent No. 765935, granted the writer, July 26, 1904, and that the multiple discharge openings with such a pump have been patented by Sulzer. Such pumps are used mostly for sinking pumps in mining services.

There seems to be a general opinion that the centrifugal pump can handle any kind of water satisfactorily. This, however, in the case of high pressure, high efficiency turbine pumps, is not so, as these machines have to be accurately constructed and highly finished in all parts, and any dirt or foreign material which may enter the pump can only result in wear and decreased efficiency, although pumps of special composition can be constructed to withstand the action of acid mine water. There is no similarity between the dredging pump and the high efficiency turbine pump. Nothing but failure can result from attempting to use the turbine pump as a dredging pump.

The writer is naturally an exponent of the centrifugal pump, but there are some conditions for which he would never think of recommending a centrifugal pump, if it could be avoided, and these conditions are just the ones which the author of the paper has selected to illustrate what can be done with the centrifugal pump. He proposes to pump twenty million gallons per day against a head of 185 feet with a 15-30 × 16 compound engine. The work which he proposes to do requires about 648 water horse-power, and he proposes to do it with an engine which, at a very liberal rating, would not deliver over 500 horse-power. To do this work would require about a 1000 horse-power engine, and, as the author of the paper states, about 90 to 100 million duty would be all that could be expected from the outfit. Such an engine and centrifugal pump would cost as much as a high duty pumping engine, which would give a duty of about 150 million foot pounds per thousand pound of steam. Thus it can be readily seen that there is not much of a field for the centrifugal pump in this line until the steam turbine driven centrifugal pump

is brought to a degree of perfection which will compare more favorably with the high duty pumping engine.

It is, however, an entirely different proposition when hydraulic generated electric power is available, for then it is possible to obtain the high speed necessary to make an efficient and inexpensive pump and motor. The cost of operation will also compare favorably with any steam-driven plant. Such an installation as this can be seen operating continuously at Schenectady, New York, and in the near future there will also be one in operation at Buffalo, N. Y.

The writer will have to admit that he has been unable to make head or tail out of the formulas for speed which the author of the paper has worked up from the old formula of Appold. Nor can he understand the diagrams illustrating the results from these formulas; neither has the writer been able to obtain any results which would even approximate what would be expected by inserting various values in the formula given, and believes that it has not been correctly stated in the paper. Appold's formula simply being an empirical one, the results which it gives could be expected only to approximate those actually obtained from the ordinarily designed centrifugal pump, and cannot be considered accurate enough for modern requirements. There seems a small excuse for using such formulas at this late day, owing to the fact that the correct theory of the centrifugal pump is well understood by the leading engineers engaged in this work.

The efficiencies shown on the diagrams and in the tables cannot be considered exceptional, as they have been greatly exceeded in this country and Europe.

Mr. C. C. Wilson.—Mr. Webber must be in error when he speaks of the value of $\sqrt{2gh}$ being equal to some number between four and five hundred, as this would be true for only one value of h, the head, which would be greater than twenty-five hundred feet.

Now take h equal to the suction and discharge heads, plus friction head of suction and discharge pipes. Then, if the peripheral speed of a runner be made equal to $1.1\sqrt{2gh}$ feet per second, the results will be satisfactory when heads vary from 30 to 60 feet, and vanes of runner are curved back to make a small angle with tangent to runner.

Or $$\text{R. P. M.} = \frac{2017\sqrt{h}}{\text{diam. of runner in inches}}.$$

It is well known that the straighter the vanes of a runner the fewer the revolutions required to obtain a given head, so no formula so simple as those quoted would fit all cases.

Unless the formula took account of friction in runner and casing as well as delivery head and angle of vanes, it would not be suited without change to widely varying conditions.

Mr. C. B. Rearick.—Referring to Mr. Webber's paper, paragraph 16, concerning balance against end thrust, our experience has shown this to be a very desirable feature, if not essential for good results in service. The design should also insure perfect balance at all suction and discharge pressures to which the pump is subject.

As paragraphs 25 and 26 mention a probable duty of 90,000,000 with a compound engine, presumably condensing, it may be interesting to note the actual results of a 24-hour duty trial of a De Laval steam turbine-driven centrifugal pump, made April 21st and 22nd, 1905, at the Mt. Prospect Pumping Station, Brooklyn, New York. The pump was designed for 6,000,000 gallons in 24 hours, and for 175 feet total head. The turbine for 150 pounds steam and 28 inches vacuum, no superheat.

The turbine is of the regular De Laval simple wheel type, running at 10,500 revolutions per minute, and gearing down to 900 revolutions per minute for the pumps.

The pumps of the series type; that is, two pump cases, each provided with double intake balanced impellers, arranged to form a two-stage pump.

Test was started at 4 p.m., April 21st, and completed at 4 p.m., April 22nd, and conducted by Mr. R. P. Bolton, expert for the city.

All readings were taken every ten minutes. All steam passing through the main pump and auxiliaries was condensed and weighed together with drips. The discharge from the pump was measured by a Venturi meter. All gauges were checked for accuracy by use of a mercury column, after the test, and the results corrected to correspond. The figures are, therefore, quite trustworthy. The results, as reported by Mr. Bolton, are as follows:

Date of test	April 21–22, 1905.
Duration of test	24 hours.
Speed of pump runners (impellers)	907½ r.p.m.
Water from condenser (steam used by turbine pump)	104,551.50 lbs.

Turbine drips	1,240.50 lbs.
Heater drips (steam used by air pump)	2,180.00 lbs.
Total condensation	107,972.00 lbs.
Percentage of moisture in steam	$1\frac{11}{100}$ per cent.
Dry steam supplied to turbine and auxiliaries	106,773.51 lbs.
Steam pressure (per sq. in.)	145.83 lbs.
Vacuum	28.84 in.
Heat in steam	127,360,084,323 B.t.u.
Feed water heated—3402.252 cu. ft. @ 61.843 lbs.	210,405.47 lbs.
Temperature of feed water entering heater	56.078 ° F.
Temperature of feed water leaving heater	113.343 ° F.
Increase in temperature	57.265 ° F.
Heat returned in feed water	12,073,276.274 B.t.u.
Net heat charged to apparatus	115,287,439.266 B.t.u
Discharge head (per sq. in.)	79.496 lbs.
Suction head (per sq. in.)	5.157 lbs.
Net head (corrected for difference in gauge levels)	173,469 ft.
Water pumped—6,546,899 gal. @ 8.34 lbs.	54,601,137.66 lbs.
Work done (foot pounds)	9,471,604,748.842
Duty (foot pounds per million B.t.u.)	82,156,433.33

Referring to the above it will be found that the duty per thousand pounds of steam, including all auxiliaries, is 88,707,440. We can also deduct the following from the above:

Dry steam supplied to turbine pump only	103,391
" " " " " " including turbine drips	104,631

From these appended data it will be seen that the duty of the turbine pump, exclusive of drips, is 91,609,567, and that the duty, including drips from the turbine, is 90,523,866.30518.

It will be noted that a 28.8-inch vacuum was maintained in the turbine case. While this was probably somewhat in favor of the turbine, the machine was nozzled for $27\frac{1}{2}$-inch vacuum, and consequently did not get the full benefit of the great increase in vacuum over what the machine was designed for. With 200 pounds steam pressure, 28 inches vacuum, and 100 degrees superheat, from our experience for the turbine end of the machine only, the efficiency would be bettered in practice such an amount that the duty would figure 103,000,000. It will be noted that this is a much smaller pump than the imaginary pump cited by Mr. Webber.

Mr. H. H. Suplee.—I would like to say a word about the pumps to which Mr. Webber refers, built by Mr. Cooper. I had the pleasure of examining those pumps at the time they were built at the Southwark foundry in Philadelphia. Mr. Cooper was there at the time, and I remember that he spoke

particularly of the difficulties that he had had with them. He said there was very little available literature on the subject, and that he had to work out every detail from the beginning. The cases of these pumps were so large that he was afraid he would have to cut them in two in order to ship them out to California by rail, or else send them around all the way by water. Finally they got a long coal car and placed the two cases on it side by side, and, by cutting the bottom of the car out, and letting the cases down as far as they could, they were enabled to ship them by rail, as they just cleared the low bridges and the tunnels. These pumps were a great success.

There is one point that I would like to call attention to in regard to the shape of the runners in the centrifugal pump. There have lately been a number of experiments made by James Alexander Smith, of Melbourne, Australia, using a miniature pump with glass sides to the case, projecting the image on a screen with a stroboscopic apparatus so that the moving parts can be made to appear stationary and the flow of the liquid demonstrated with different-shaped runners, and in that way the most effective shape found. He has published two papers, to be found in the *Transactions* of the Victorian Institute of Engineers, and his conclusions were that a very nearly radial vane gave the best results. These papers are on record and can easily be consulted.

Mr. D. J. Lewis, Jr.—I might say, Mr. President, that I was associated with Mr. Cooper at the time he built those pumps for the Mare Island Navy Yard, and what bothered him most was the guarantee that the government required, which was that after the pumps were tested, if they did not come up to the requirement they would have to be taken out and the plant restored to its original condition. So he made his test on the drainage pump, which was a 16-inch pump, and he found that he was about 25 per cent. over the guarantee.

Pumps just like those can be seen in the Simpson dock at the Brooklyn Navy Yard, in the Navy Yard at Portsmouth, Va., the League Island Navy Yard, and in the first dock of the Newport News Shipbuilding Company—all of the same pattern.

Mr. Rice.—This paper gives the present state of the published mathematics of the centrifugal pump very clearly. I have tried to find some mathematical treatment of this subject in published form, but without success. A rational mathematical treatise on centrifugal pumps would undoubtedly lead to improvements

in design, which would in turn lead to increased duty. At the present time the low duty obtainable from these pumps forms an insuperable obstacle against centrifugal pumps for high duty service. When this obstacle is overcome centrifugal pumps driven by turbines will form an ideal form of pumping engine, and should take the field to the exclusion of other forms of pumping engines. The fact that the centrifugal pump has only one moving part, that it has no valve, and that it is extremely simple and low in cost, will, I think, sufficiently demonstrate its possibilities.

The author states that the form of curves used for his impeller plates is extremely efficient, but the efficiency curves given do not seem to bear this out. Efficiency curves published by other experimenters have shown equally good results.

Mr. W. H. Morse.—There is one phase of this subject that has not been touched upon, and that is the possibility of combining a centrifugal pump with a gas engine, and thus obtaining an efficiency which is fairly comparable with a high duty pumping engine. That can now be accomplished. The efficiency of centrifugal pumps is now demonstrated and guaranteed as at least 65 per cent. Indeed, I know pumps that are giving an efficiency of 74 per cent. in Schenectady. When I was in Australia last year I heard favorable reports of pumps in South Africa which were reputed to be giving an efficiency of 80 per cent., although I do not vouch for them. Assuming, however, that the efficiency is 65 per cent., and that you can connect one of these pumps to a gas engine, we can obtain a duty of between 120 and 130 million foot-pounds per one pound of coal, and if we assume 75 per cent. efficiency, we can obtain a duty of 145 to 150 million. I realize that high duty pumping engines have exceeded that duty, but I do not think you will find that they will hold up in actual practice even to that. Moreover, it will be found that the cost of the plant of a high duty pumping plant, as compared with a gas engine, installation, will be considerably greater. By that I mean, if you compare the cost of boilers, stacks, settings, and high duty engines with the cost of the gas producer, holder, the gas engine and the centrifugal pump, you will find that the costs of the machinery is about one-third less. I am referring to a plant that I figured on recently for about 20,000,000 gallons per 24 hours, and a head of 250 feet. Such a plant is to be particularly considered in lo-

calities where there is a fuel not well adapted for steam raising, as for irrigation in the West, and the tests made at the St. Louis Exposition have shown that good results can be obtained by gasifying poor Western coal.

The President.—Perhaps in Scranton it might be well to suggest that these culm heaps might be utilized in that way.

Mr. L. A. Zohe.—I would like to ask the last speaker who is the manufacturer of the pumps that give that high efficiency? I know of an installation where a guarantee of 65 per cent. was offered, but the efficiency was never reached.

Mr. Morse.—I think that the Worthington Company will guarantee an efficiency of 65 per cent. for pumps on any reasonable head. I will say that these figures were not given to me by the Worthington people.

Mr. Zohe.— My modesty will not permit me to mention names, but I think it is not practical for centrifugal pumps for actual use to figure very much over 60 per cent. I know some manufacturers give 75 per cent., but I think practical demonstrations very rarely go over 60 per cent., and they are more likely 40 per cent.

Mr. Morse.—About six years ago I tested some pumps which are now in the filtration plant at Albany built by the Lawrence Machine Company. They were guaranteed for 65 per cent. efficiency, and they actually gave that percentage on tests.

Mr. Mattes.—Has the development of the multiple stage centrifugal pump reached that point yet where a guarantee is given regardless of the head? My own understanding, without any experience with other than single stage centrifugal pumps, has been that after the head passed a critical point the curve of efficiency drops, and with an ever increasing rate of decrease. Another point that I should like to call attention to is that Mr. Morse's remarks largely refer to the power ends and not to the pumps. It was in large part an argument in favor of the gas engine; but we can use the gas engine to propel reciprocating pumps as well as rotary pumps. It seems to me that is a matter entirely separate from the question of pumps.

The President.—Mr. Rice has referred to the mathematics of the centrifugal pump. I made some studies and experiments on centrifugal pumps about ten or fifteen years ago, and came to the conclusion that the mathematics, when they were finally worked out for the practical constructor, would be of a somewhat simple nature, and that the problem would be very much that of the

turbine. Those of you who have studied the turbine question know that there has been no end of complicated formulæ worked out for determining the best form of turbine bucket, but, nevertheless, the best turbines have been built not so much from the curves of mathematicians as by the pattern makers under the guidance of men who had a sense of the fitness of things and of forming their channels and water ways so as to avoid eddies and shock to the current. I think it will be very much the same with the centrifugal pump, that the necessary mathematics are of a simple order and relate mainly to the fundamental propositions, and that the main thing is to keep the fluid friction in the space between the vanes and the case at a minimum and to shape the vanes at their extremity so that as far as possible they throw the water back, depositing it as nearly as possible in a condition of rest, very much as is an essential feature in the discharge of a turbine duct. And just as our best turbines have been built, not by mathematicians, but by ingenious, thoughtful mechanics, repeatedly cutting and trying and testing in such a place as the Holyoke testing flume. I think the centrifugal pump of the future will be perfected by a testing flume such as Mr. Webber began with at Lawrence, and designing the shape of the buckets or vanes by a process of "cut and try," under the supervision of someone who has a full command of the simple fundamental theory of the flow of water through passages, and a conception of shaping the passages so that a minimum of eddy will be produced.

*Mr. William O. Webber.**—Replying to Mr. Henshaw, would say that I have obtained 65 per cent. efficiency with careful tests on a 6-inch pump, and 47 per cent. efficiency with as small as a 2½-inch two-stage pump, pumping against 53 pounds pressure. This would lead us to believe that we can get at least 70, and possibly 75 per cent. of efficiency on multiple stage pumps of 15 or 20-inch discharge.

I know that centrifugal pumps have been used for pumping oil and kerosene, but should not think that they would be applicable to oil pipe lines for the reason that the flow in these lines must be necessarily at a very low rate in order to obviate skin friction.

The Mississippi River Commission, in "Reports of Efficiency

* Author's Closure, under the Rules.

Tests of Hydraulic Dredges," January 23d, 1904, reports many tests, some as high as 80 per cent. efficiency of pump alone, and a combined efficiency of pump and engine of 74 per cent. These tests are with steam at from 130 to 150 pounds pressure, 23 to 24 inches of vacuum, speeds from 125 to 160 revolutions per minute. The pumps were large pumps of 32 to 34 inch discharge, with impellers from 5 feet 9 to 7 feet in diameter, the suctions varying from 5 to 20 feet, and discharges varying from 12 to 40 feet.

They draw the conclusions that "the most efficient speed for centrifugal pumps, for the size under consideration, is one giving a peripherical velocity of about 50 feet a second, and that the most efficient head is from 45 to 50 feet; that the closed, or shrouded, runner, while being a necessity to prevent abnormal wear of pump casings and runner (in dredging pumps), and thus increasing the clearance and decreasing the efficiency very rapidly, is also somewhat more efficient than an open runner, even with a small clearance, the tests showing from two to four per cent. in favor of the enclosed runner; that the tapering blades on the runner and curved vortex chambers are slightly more efficient, but that the additional expense of construction and maintenance is hardly warranted by the slight gain; that the area of the opening in the suction head and discharge pipes should be approximately the same in order that the velocity may be maintained at a nearly constant rate."

The writer was very glad to see Mr. Rearick's paper upon "Tests of De Laval Steam Turbine," at the Mt. Prospect Pumping Station, Brooklyn, N. Y. I have no doubt that such results can be duplicated.

Replying to Mr. Wilson and Mr. Ray. The writer is pretty thoroughly familiar with the theory of centrifugal pump construction and practice. The only trouble is that the practice and accepted theory do not coincide. If Mr. Ray has any theoretical formulas which will fit modern practice, he will benefit the profession at large by publishing them.

The writer has made many tests and experiments on all methods of balancing centrifugal pumps, shown in the different cuts, and described in the various patents issued in this country and abroad, and has failed to find any method of balancing comparable with that shown in the accompanying cut, on which the writer was allowed a United States Patent on June 6th.

A multiple stage pump, with a plurality of inlets, is shown in a United States Patent to P. Kugel, V. Gelpke, November 3d, 1903, so that Mr. Ray cannot claim that feature.

Sulzer's American patent does not broadly cover the multiple discharge openings, but only in combination with multiple inlets.

The writer has also been allowed a United States Patent on an improved ring oiling shaft bearing, in connection with centrifugal pumps, which, while removed from contact with the water being pumped, does not require a separate bearing to be kept in line.

Mr. Ray's criticism as to the high duty service, for which he would never think of recommending a centrifugal pump, is pretty completely answered by Mr. Rearick's paper, showing such a pump doing high duty work. The future will alone show what can be done in these lines.

The writer is sorry that he omitted to give Mr. John H. Cooper due credit for the work he did in constructing the Mare Island Navy Yard pumps. It had entirely slipped his memory.

In reply to Mr. W. H. Morse, of New York, would say that the writer is experimenting, and has already constructed a combined centrifugal pump and gasoline engine for contractor's use, and hopes, before long, to make some tests of a gas producer and gas engine directly connected to a centrifugal pump.

Replying to Mr. Mattes. The writer's tests would show that the efficiency of a single-stage pump drops after passing a lift of from 35 to 40 feet. Tests which he has made on multi-stage pumps would lead him to state that a guarantee of efficiency could be given with a multi-stage pump, almost regardless of the head, if the question of first cost was not taken into consideration.

The commercial side of the question is a very vital one, but some really remarkable results could be guaranteed with commercially possible pumping units.

The writer fully agrees with the remarks of our President as to how the ideal centrifugal pump will be perfected.

This paper was largely submitted in hopes that others would follow, stating the theories which have best stood comparison, with practical results.

No. 1092.*

STEAM ACTUATED VALVE GEAR.†

BY W. H. COLLIER, JACKSON, TENN.

(Member of the Society.)

1. Notwithstanding the high engineering talent engaged in engine design and the vast number of improvements made in recent years, it still appears to be possible to greatly improve the present types of reciprocating engines when efficiency and construction are considered. Attempts to secure a perfect distribution of steam by mechanically operated valve gears have necessarily resulted in complicated mechanism and excessive friction in these parts and usually in limits to the rotative speed of the

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* Presented at the Scranton meeting (June, 1905) of the American Society of Mechanical Engineers, and forming part of Volume XXVI. of the *Transactions*.

† For further discussion on this general subject consult *Transactions* as follows:

No. 31, vol. 2, p. 124: "Brief Treatise on the Steamboat Cam." Lewis Johnson

No. 97, vol. 4, p. 150: "Back Pressure on Valves." S. W. Robinson.

No. 116, vol. 4, p. 268: "Balanced Valves," C. C. Collins.

No. 130, vol. 5, p. 34: "Motion Curves of Cut-off Valves." A. Wells Robinson.

No. 335, vol. 10, p. 521: "Cornish or Double-Beat Pump Valves." A. F. Nagle.

No. 567, vol. 15, p. 177: "Buckeye Engine Valve Gear." A. K. Mansfield.

No. 616, vol. 16, p. 117: "Cam for Actuating Valves of High Speed Steam Engines." Charles T. Porter.

No. 617, vol. 16, p. 134: "Improved Centrifugal Governor and Valve." Charles T. Porter.

No. 769, vol. 19, p. 449: "Stevens Valve Gear for Marine Engines." Andrew Fletcher.

No. 793, vol. 20, p. 42: "Valve Gear of the Willans Engine." John Svenson.

No. 809, vol. 20, p. 475: "Allen Valve for Locomotives." C. H. Quereau.

No. 824, vol. 20, p. 967: "New System of Valves for Steam Engines, Air Engines and Compressors." F. W. Gordon.

No. 920, vol. 23, p. 151: "New Valve Gear for Gas, Steam and Air Engines." E. W. Naylor.

No. 1017, vol. 25, p. 138. "Improvement of Valve Motion of Duplex Air Compressor." S. H. Bunnell.

No. 1022, vol. 25, p. 2121: "Construction and Efficiency of a Fleming Four-valve Engine Direct Connected to a 400 kw. Generator." Benj. T. Allen.

engine, while the point of cut-off in most cases is limited to one-half stroke with a constant point of compression.

2. These objectionable features, it is believed, have been overcome by the engine described in this paper and, aside from simplicity of construction and increased economy due to high rotative speed and minimum clearance, the following advantages are claimed:

a. Perfect distribution of steam. All movements of valves are full and instantaneous, the cut-off varying under control of governor from no load to full load, with variable compression.

b. Closer regulation. The governor has absolutely no work to perform, but merely indicates the point of cut-off.

c. No safety device is necessary, as the speed limit is governed by the velocity of the steam through the admission port of the governing valve, which makes it absolutely impossible for the speed of the engine to increase 50 per cent. above that for which it is designed.

d. A higher rotative speed of the engine. The speed of the engine is limited by piston speed only.

e. The engine is reversible. It may, by means of a sliding key, be started, stopped or made to run in either direction.

f. A valve gear in which it is impossible to get the valves out of adjustment.

g. A more economical valve gear. It does away with friction and weight of releasing gear, dash pots, rocker arms, eccentrics, etc., the steam on the actuating pistons being used expansively.

3. The following is a description of a 14 by 30 engine running 100 revolutions per minute, developing from 80 to 120 horsepower, which has been in operation ten hours per day for the past six months in the power plant of the Southern Engine and Boiler Works, and has verified the above claims.

4. The body or frame A of this device is connected to the side of the cylinder as shown in Figs. 286, 287 and 288.

5. The controlling valve B is connected to the gears J and J_2 by a sliding key Q, and in such a way as to engage either of the gears J or J_2, or it may occupy a central position, in which position the gears are free to move on the sleeve K. The shaft L, which is fixed to the controlling valve B, is connected to that sleeve K by a sliding key, which causes the controlling valve to revolve with the sleeve K but allows a free lateral motion. Lateral motion is imparted to the controlling valve B by a bell crank M, which is connected to a

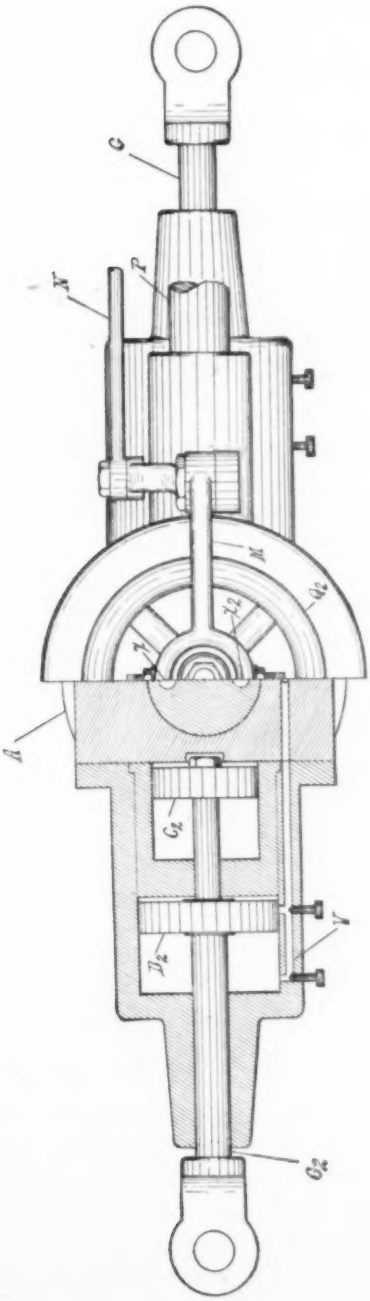

FIG. 286.

governor by the reach rod N. The gears J and J_2 are caused to revolve by the gear O, which is connected to the shaft of the engine by the shaft P and gears. The gears are so proportioned as to cause

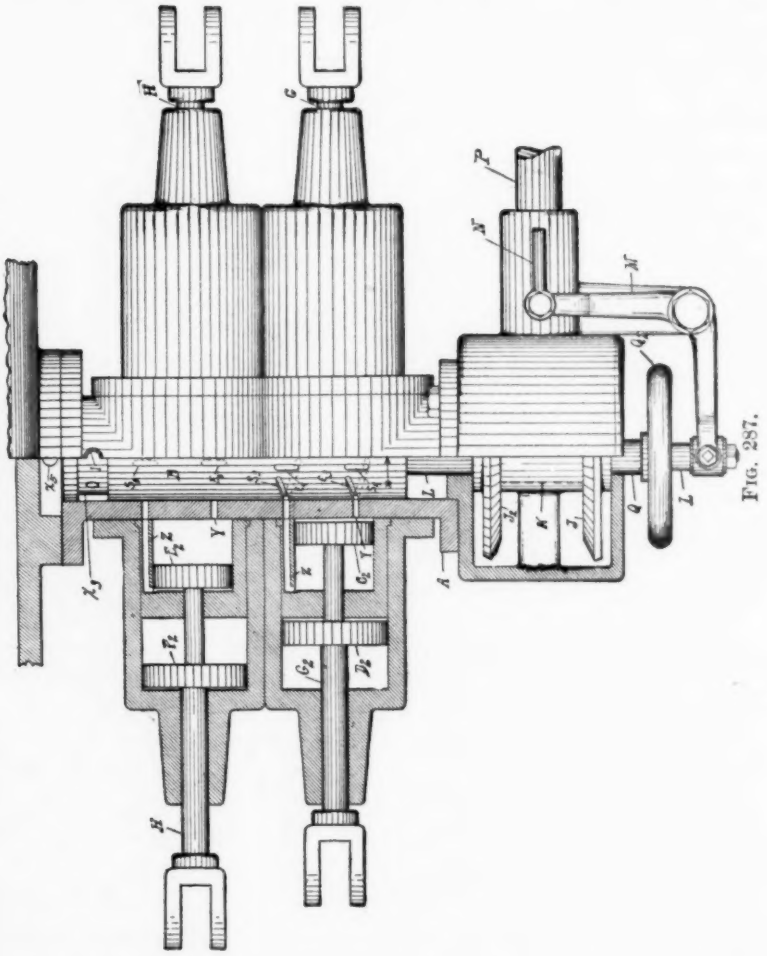

FIG. 287.

the main shaft of the engine and the controlling valve to make the same number of revolutions. The two gears J and J_2 are for the purpose of reversing the engine. The valve revolves in the direction indicated by the arrow, and if it is desired to run the engine over, the controlling valve is connected to the gear J, which causes the controlling valve to revolve in the direction indicated by the

arrow; if the engine is to run under, the controlling valve is connected to the gear J_2, which causes the engine to run in the opposite direction while it revolves the controlling valve B as indi-

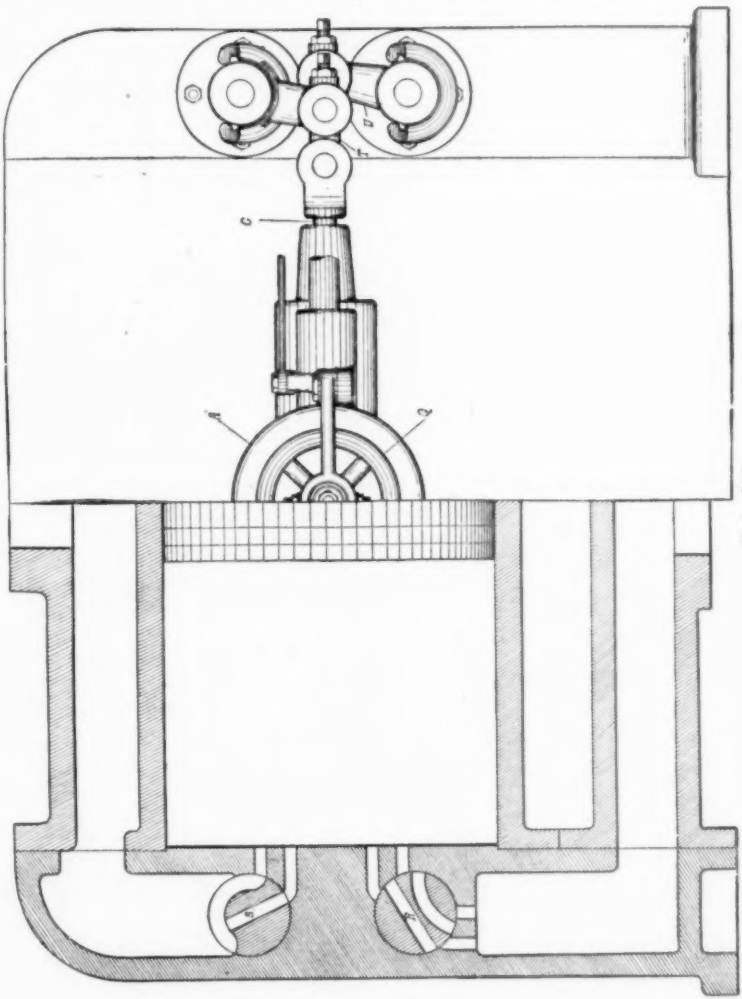

FIG. 288.

cated by the arrow. The slots for the sliding key Q in the gears J and J_2 are so located as to give the proper lead to the engine.

6. There are four actuating pistons, C, C_2, E, E_2, which are connected to the steam and exhaust valves of the engine by means

of the rocker arms U (Fig. 288). The large pistons, D, D_2, F, F_2, are for the purpose of cushioning the actuating pistons, C, C_2, E, E_2, in their extreme movements. The ports V are for the purpose of allowing a free movement of the pistons D, D_2, F, F_2 except at their extreme points of movement.

7. The controlling valve B has a series of oblong and helical grooves which are connected either to the exhaust passage X_2 or to the steam passage X by means of ports not shown.

8. The steam passage X is connected to the steam pipe I by means of the annular groove X_3. The actuating pistons C, C_2, E, E_2 have ports Y, Z at the end of their cylinders.

9. The exhaust passage X_2 extends into an opening in the side of the cylinder, which allows the exhaust to pass out through the opening X_5 to the main exhaust passage of the cylinder or to the atmosphere (Fig. 287).

10. The helical grooves S_3, E_3 control respectively the steam and exhaust passages of the cylinders, which contain the actuating pistons C, C_2, and regulate the point of cut-off, the controlling valve revolving in the direction indicated by the arrow, the point of cut-off varies as the lateral movement of the controlling valve; this lateral motion is caused and governed by a governor connected to the reach rod N, and as the controlling valve moves outward from the cylinder the helical groove S_3 comes more quickly into contact with the ports Z, moving the actuating piston inward, closing the admission valve of the engine, while at the same time the helical groove E_3, which is connected to the exhaust passage X_2, uncovers the port Y of the piston C, allowing the steam in that end to exhaust through the passage X_2.

11. The oblong grooves S_4, E_4 control the actuating pistons C_2 opening the admission valves of the engine, they being parallel with the movement of the controlling valve are not affected by the lateral movement of the same; that is, the point of admission remains constant. The same is true of the oblong grooves E_5, S_5, which are on the opposite side of the controlling valve, and which control the actuating pistons E, E_2, which operate the exhaust valves of the engine during release. The actuating pistons E, E_2 operate the exhaust valves of the engine during compression, and are controlled in this movement by helical grooves S_6, E_6 not shown.

12. The grooves S_3, S_4, S_5 and S_6 are connected to the steam passage X, and the grooves E_3, E_4, E_5 and E_6 are connected to the exhaust passage X_2.

13. The outlets of ports Y, Z of the cylinder, which contain the piston C, are diametrically opposite those of the cylinder containing the piston C_2 and in the same vertical plane. The same is true of the outlets of the cylinders containing pistons E and E_2.

14. Assume the engine to be rotating in the direction indicated by the arrow head and to be on dead center. At this instant the oblong passage S_4 is admitting steam on the inner face of the actu-

FIG. 289.

ating piston C_2 through the port Y, opening the admission valve at this end of the cylinder, while the oblong passage E_4 has opened the exhaust passage Z at the opposite end of C_2. At the same instant the oblong groove S_5 has uncovered the port Y, forcing the actuating piston E outward, opening the exhaust valve at that end of the cylinder, the oblong passage E_5 uncovering the exhaust port Z at this instant. The engine and controlling valve B will continue to revolve until the helical grooves S_3, E_3 uncover respectively the steam and exhaust ports of the cylinder containing the actuat-

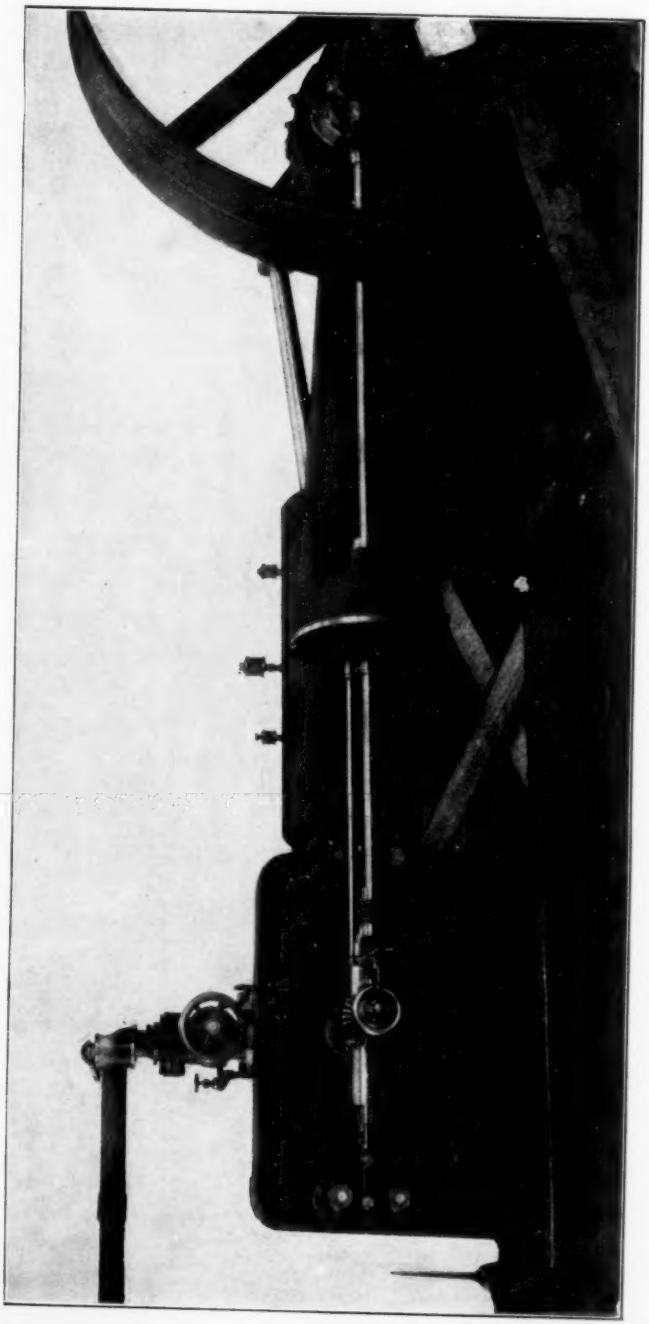

FIG. 290.

ing piston C_2, causing the piston to move inward, cutting off the steam at this end of the cylinder. The engine and controlling valve continue to revolve until the helical grooves S_6, E_6 uncover the openings Z, Y, respectively forcing the actuating piston E inward, closing the exhaust valve at that end of the cylinder, causing the stem to be compressed until the end of the stroke is reached, when the engine starts on the return stroke the movements of the valves are reversed until the engine completes its cycle.

15. It will be observed that as the speed of the engine increases the controlling valve B (Fig. 287) is forced outward, that is, away from the cylinder by means of the governor connected to the reach rod N, bringing the helical grooves S_3, E_3, S_6, E_6 nearer their respective ports, which causes an earlier movement of their respective actuating pistons. That is, as these helical grooves are forced out-

FIG. 291.

ward they approach more closely their respective ports, requiring a smaller angle of movement in uncovering the same. In the position shown (Fig. 287), the grooves S_3, E_3 would not uncover the ports of the cylinder containing the actuating piston C_2, operating the cut-off at that end of the cylinder until almost half the stroke is made; if, however, the controlling valve is forced outward by means of the governor, cut-off will take place earlier and earlier until the ends of ports S_3, E_3 are in the same vertical plane Z, Y (Fig. 287), in which case cut-off will take place on the instant after admission, and while the engine is practically on dead center.

16. To start the engine from a state of rest, the pin Q is withdrawn until the sliding key is in its central position; that is, occupying the space between gears J, J_2, which allows the sleeve K, and with it the controlling valve B, a free rotary movement, which may be imparted by means of the hand wheel Q_2 operating the steam and exhaust valves of the engine. The hand wheel is revolved in the direction indicated by the arrow, which leaves the valves of the engine in the proper position for starting. The key is then pushed inward or outward, depending upon which direction

the engine is to run, engaging J, or J_2, in which position the engine will continue in motion.

17. Should it be desired to stop the engine instantly, the pin Q is withdrawn until it occupies the space between gears J, J_2, disengaging the sleeve K from said gears, and allowing the controlling valve B to come to a state of rest, stopping the engine almost instantly.

18. The limit of the speed of the engine is governed by the width of the slot S_4 of the controlling valve B, and this slot controls the admission pistons C, C_2. The width may be so proportioned that at the required limit of peripheral speed of the controlling valve the steam will not have sufficient time to operate the admission pistons C, C_2, that is, the width of this port might be sufficient to allow the required amount of steam to operate the actuating pistons at a certain number of revolutions and would not allow sufficient steam at an increase of 50 per cent. above this speed.

19. A gear embodying the above features is now being applied to an air compressor, and it is believed that its application to compressors is even more desirable than to engines. The above principles may be applied to any stationary engine or locomotive having two or more valves.

DISCUSSION.

Mr. F. M. Rites.—The description of this new mechanism is very interesting, especially as the writer of the paper seems to have positively simplified the tripping mechanism of the Corliss engine, while actually increasing the efficiency of the combination.

It would be still more interesting if the writer of the paper could give an actual indication of the practical limit of speed of operation of an engine so equipped, and I should like to ask if he has yet had time to determine such limits, and also at what speed the diagrams from the engine become objectionably affected.

It seems certain from the simple character of the mechanism that the available speed should be considerably higher than the usual releasing gear.

Generally speaking, it would seem a distinct advance in the art of engine design.

Mr. Suplee.—I would like to ask if any member has any information about the valve gear of the old Babcock-Wilcox engine?

My impression is that it had a steam thrown valve gear; that is, I think there were flat slide valves driven by an eccentric and a riding cut-off valve on top of that, which was steam thrown. I think our past President, Mr. Babcock, spent a great deal of time on it, and in tests with the Corliss engine obtained very good results.

Mr. Kent.—I think that the test referred to by Mr. Suplee will probably be found in the published *Transactions* of the American Institute (New York City). The test was made at the American Institute Fair in 1871, I believe, when that engine was exhibited in competition with other engines. It is quite possible that the record of the test will also be found in some of the scientific publications of the time, such as the *Scientific American.*

I trust that before the paper is finally printed in the *Transactions* the Secretary will communicate with some of the members who are specially expert in this line, such as Mr. Stanwood, of Cincinnati, and ask for a more complete discussion than it is possible to give to the subject here. I think this whole subject is important enough to be threshed out well and not passed over with such very brief discussion as we can give it to-night.

Mr. Richard H. Rice.—I am unable to agree that this valve gear will be as useful and beneficial as the paper would seem to indicate. I am inclined to think that it will be found in a short time to wear leaky, and will then be not so reliable as a gear which is driven directly from the shaft of the engine, and which either has to move with it or cause a rupture of some part. In this latter case the question of the operation of the gear is simply a question of size and strength of the parts. In the gear in question we have pistons which move under the influence of various disturbing causes, such as the lack of lubrication and the possibility of the introduction of foreign matter, and my experience with gears actuated by other means than mechanical ones would lead me to expect that it would have a short life.

*Mr. W. H. Collier.**—The objectionable feature of this valve gear as pointed out by Mr. Rice, that is, "not so reliable as a gear which is driven directly from the shaft of the engine and which either has to move with it or cause a rupture of some part," is one of the advantages claimed for this gear.

Mr. Rice states that the "lack of lubrication and the possibility

* Author's Closure, under the Rules.

of the introduction of foreign matter are also objectionable features." In this connection I will state that the steam supply for this valve gear is connected to the steam pipe of the engine below the lubricator giving the gear the same lubrication as the engine. An individual lubricator is also provided for this gear. The above method of steam connection precludes the possibility of foreign matter entering the valve gear.

A recent examination of this valve gear after same had been in daily operation for eleven months, three months of that time for twenty hours per day, shows that the wear on the gear is not perceptible. I would further state that the valve gear has not received repairs of any kind or caused one moment's trouble during this entire time.

I would also add that in a heavy duty engine of this type, completed two months ago, the cushioning cylinders have been eliminated and the actuating pistons made to cushion on exhaust steam while the ports of the actuating cylinders are beneath the controlling valve insuring a steam-tight valve. This engine was made to run 210 revolutions per minute, the strength of the fly wheel preventing a greater speed.

No. 1093.

FRANZ REULEAUX.

IN MEMORIAM.

BY. H. H. SUPLEE, NEW YORK, N. Y.

(Member of the Society.)

Professor Dr. Franz Reuleaux, Honorary Member of the Society, died at his home in Berlin, August 20, 1905.

Prof. Reuleaux was born September 30, 1829, at Eschweiler, near Aachen, Germany, so that he had nearly completed the seventy-fifth year of a life of activity and productiveness devoted to the advancement of the applied science of engineering. His father, Johann Josef Reuleaux, was the founder of one of the earliest machine shops in Germany, and in this establishment the youth acquired practical shop experience until attracted by the growing fame of Redtenbacher, then professor of machine design at the Karlsruhe Polytechnic School, Reuleaux entered that institution in 1850. It is thus interesting to note that the future professor and author acquired his shop experience first and followed this with the best technical course then available; and, as if this was not enough, he proceeded to the universities of Berlin and Bonn, thus securing a literary and philosophical education as well as a practical mechanical training.

While a student at Bonn, and in collaboration with his friend C. L. Moll, Reuleaux issued his first work, entitled: "*Konstruktionslehre für den Maschinenbau*," this being followed by another, also in association with Moll, "*Die Festigkeit der Materialien.*" These two works, treating respectively of machine design and of the strength of materials, show marked originality in the discussion of the subjects, being the first in which the modern notions of the elastic theory and the working fiber stress appear in distinction from the old method of employing only the ultimate resistance of the material tempered by an assumed factor of safety. From the very first Reuleaux avoided the use of arbitrary and

empirical methods, and strove to impress the importance of accurate and precise knowledge of the actual conditions of work.

In 1854 Reuleaux left the university of Bonn and went to Cologne, taking charge of a machine shop there, and thus adding to

FIG. 292.

his practical experience, but in 1856 he entered upon his true career as a teacher by becoming Professor of Mechanics at the Zürich Polytechnikum, this institution having just been opened. Here he formed one of a group of famous engineers and instructors, among whom, as original contributors to knowledge, may

be also mentioned Zeuner, Culmann and Clausius, so that from this earnest little Swiss town, the home of Pestalozzi, and always foremost in the science of education, emanated the modern theory of machine design, the development of the mechanical theory of heat, the evolution of the slide-valve diagram, and the modern science of graphical statics.

At Zürich Reuleaux wrote his treatise: "*Berechnung der Wichtigsten Federarten*," this being a treatise upon the proportions of springs, a subject about which there had until then been no definite information upon record, and in preparation of which he had to undertake an exhaustive series of experiments, the models for which may yet be seen in the cabinet at the Zürich Polytechnic. This work was followed by the first of his great treatises, "*Der Konstrukteur*," a book which has had a controlling influence, acknowledged and unacknowledged, upon machine design in all parts of the world.

The first edition of this book appeared in 1861, and succeeding enlarged editions appeared in rapid succession, the fourth, nearly three times the size of the first, being published in 1889, and the work having been translated into French, Swedish, Russian and English.

While at Zürich Reuleaux received a call to the Polytechnic School at Riga, of which the scope was then being extended; this he declined, but shortly afterward, in 1864, he accepted a call to become professor at the *Gewerbe Akademie* in Berlin, and when this institution was consolidated with the Royal Academy of Architecture to form the great Royal Technical High School at Charlottenburg, known all over the world as the *Königliche Technische Hochschule*, Reuleaux was made its director, a position which he held until 1896, when he was retired, having completed his forty years of active professional work.

Professor Reuleaux took an active interest in international expositions, and represented Prussia in London in 1862, at Paris in 1867, Vienna in 1873, and Philadelphia in 1876, as member of the jury, and also as Imperial German Commissioner at Philadelphia in 1876, Sydney in 1879, and Melbourne in 1881, and was also Honorary Prussian Commissioner at Chicago in 1893. It was in the course of his work at the Philadelphia Centennial in 1876 that he took occasion to characterize the wholly inadequate German representation in his "*Briefe aus Philadelphia*" as "*billig und schlecht*," or cheap and nasty. That it took some courage to

antagonize the home industries in this outspoken manner is evident from the storm of disapproval which his reports awakened, but the ultimate consequence was apparent in the rapid improvement which appeared in the products of the German machine shops from that time, much of this improvement being due both to the scathing criticism which Reuleaux had had the courage to utter, and to the influence of the educational work which was being conducted under his supervision.

In 1875 appeared his most original work, the "*Theoretische Kinematik*," well known to English reading engineers through Sir Alexander Kennedy's masterly translation as the "Theoretical Kinematics."

This remarkable work is nothing more nor less than an attempt to bring the entire science of mechanical movements into a systematic form, this plan involving the invention of a new notation permitting any combination of parts to be represented by a written formula, so that, however deeply the actual mechanism might be concealed within the construction, its identity would be fully revealed simply by writing its obvious formula. Thus, in the case of a revolving crank with a connecting rod joining it to a rocker arm, the older writers considered this simple combination as composed of but three parts. Reuleaux, however, perceived that the frame or bed-plate, or even the floor, by which the axles of the crank and the rocker were held in their positions, formed a fourth member, the whole thus consisting of a closed chain or circuit, of which any member might be considered as fixed and the others free to move, and the various "inversions" of such a system led to interesting combinations. It is found that the number of such simple trains of mechanism is comparatively few, while every machine can be resolved into a combination of several trains, and it is in this manner that every variety of machine, however complex, can be analyzed. After examining the entire field of technical literature, Reuleaux failed to discover any precise definition of a machine, and hence, in his Theoretical Kinematics, he gives the following, which has not been superseded by any later author: "A machine is a combination of resistent bodies so arranged that by their means the mechanical forces of Nature can be compelled to do work, accompanied by certain determinate motions."

From the beginning of his work Reuleaux placed reliance upon the use of working models as illustrations of mechanical concep-

tions, and the cabinet of kinematic models prepared from his designs at Charlottenburg forms a noteworthy portion of the equipment of that institution. Copies of many of these models have been made for other educational institutions, the most complete being at McGill University at Montreal and at Cornell University at Ithaca.

After his retirement from the active work of instruction, Professor Reuleaux devoted himself to consulting practice and to the continuation of his literary work. The second portion of his "Kinematics" appeared in 1900, this being termed "Applied Kinematics," as supplementing the former "Theoretical Kinematics," and he also prepared numerous papers for presentation before technical societies.

While Professor Reuleaux achieved fame as an engineer and as a technical writer and instructor, he was, to those who knew him intimately, a man of broad culture and many-sided acquirements. A remarkable linguist, speaking and writing French, Italian and English freely, he was versed in literature and the arts as well as in science. While visiting the World's Fair at Chicago it was most interesting to hear his profound comments upon matters of architecture and ethnology, while a visit to the art museum showed his familiarity with details of archæology and painting. As an example of the brilliant literary capacity of the man, he visited, in 1893, the Minnehaha Falls, and this incited him to make a translation of Longfellow's poem "Hiawatha" into German poetry of a high order, this difficult piece of work being done in what was supposed to be a vacation from his professorial duties. His remarkable address, "Kultur und Technik," delivered in Vienna in 1884, and published in an English translation by the Smithsonian Institution in 1890, is another example of his broad survey of the field of art and science.

In addition to his engineering work he filled the position of a trustee of the Royal Museum of Arts at Berlin, and was a member of the Commission for the Purification of the German language. He was also honored with the appointment of Royal Privy Councillor, and was the recipient of a great number of honorary and academic degrees and titles.

Professor Reuleaux was elected an Honorary Member of the American Society of Mechanical Engineers on November 2, 1882, and contributed to its Transactions a paper upon "Friction in Toothed Gearing" at the New York Meeting in 1886.

No. 1094.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

CHARLES L. BAILEY.

Mr. Charles L. Bailey was born in Newark, N. J., March 11th, 1868. After preliminary instruction in making patent drawings he entered the drawing-room of the Pratt & Whitney Company in Hartford, 1886, and remained there for two years.

He then became draftsman for the engineering department of the municipal government of the District of Columbia.

In 1891 he owned and operated a machine shop in Minneapolis; in 1893 was designer for two years with various companies, becoming connected in 1895 with the Pope Manufacturing Company of Hartford as mechanical engineer. From 1898 to 1901 was master mechanic with the Coe Brass Manufacturing Company of Torrington, Conn., and in 1902 became mechanical engineer of the Waterbury Farrel Foundry & Machine Company, building presses and automatic machinery.

He connected himself as a junior with the Society in 1890, and was promoted in 1902.

In April, 1903, he accepted a position with the Draper Co., Hopedale, Mass., designing and developing automatic machinery for the manufacture of bobbins and shuttles; this position he held at the time of his death, which occurred at Milford, Mass., November 24th, 1904.

WILLIAM B. BOGARDUS.

Mr. Bogardus was born in Chicago in 1873, and a year later his parents brought him to Buffalo, New York. His first schooling was received at the Buffalo public schools, and later at the Buffalo High School, where he finished a four years' course in less than three years. He graduated from the mechanical engineering department of Sibley College, Cornell University, in

1896, and though only thirty-two years of age at his death, he was regarded as one of the highest authorities in the construction of Portland Cement mills, which he had made his specialty since leaving college in 1896. Upon leaving college he entered the cement business, and step by step advanced until he had become a master in his chosen specialty. He designed many of the cement mills of this country and Canada, and credit is due him for many of the improvements in the cement industry. Though he had designed a number of plants before this, his most important work was done after he had founded the Peninsular Engineering & Construction Co. of Jackson, Michigan, of which he was president and chief engineer. Included among these later plants are the National Portland Cement Co. at Durham, Ontario, The Peninsular Portland Cement Co. at Jackson, Michigan, The Southern States Portland Cement Co. at Atlanta, Georgia, and the Western States Portland Cement Co. at Independence, Kansas. All of the above were designed and built in the last five years, and aggregate in cost about five millions of dollars. Besides being employed by the above-named companies to design their plants, Mr. Bogardus was consulting engineer for other cement plants, crushing plants, and other manufacturing industries.

His death occurred at Spokane, Wash., on November 1st, 1905. He became an associate member of this Society in December, 1900.

JAMES THOMAS BOYD.

Mr. Boyd was born at Fishkill-on-Hudson, N. Y., November 25th, 1846, and was educated at the Academy in Newburg, N. Y., and in 1863 became a draftsman at the Washington Iron Works at Newburg.

In 1864 he entered the navy as third assistant engineer, and served on the S.S. "Dumbarton" until the end of the war. Soon after this he became a draftsman at the Atlantic Works at East Boston, Mass., and remained with this company until 1871, when he became superintendent of the Standard Chain Works at East Bridgewater, Mass., and the following year assisted in the design of a plant for the manufacture of pipe for the market.

In 1874 he became designing and constructing engineer for the Atlantic Works at East Boston, Mass.

In 1892 he became manager of the George F. Blake Manufacturing Co. and the Knowles Steam Pump Works.

In 1895 he left the above companies and opened an office in Boston as consulting engineer. He served in this latter capacity with the Atlantic Works, of which he was also a director, with the Edison Electric Illuminating Co. and the Metropolitan S.S. Co.

He was a member of the American Society of Civil Engineers, the Society of Naval Architects and Marine Engineers, the Boston Society of Naval Engineers, and of this Society, which latter he joined at the Pittsburg meeting in 1884.

His death occurred November 3rd, 1904, at Boston, Mass.

THEODORE F. BURGDORF.

Captain Theodore F. Burgdorf, U. S. N., was born November 22nd, 1854. He entered the Naval Academy in 1873, and after graduating became an assistant engineer in 1877. His first service was on the U. S. S. "Swatara," on which he served three years. His next duty was in Washington with the Bureau of Steam Engineering. A year later he joined the U. S. S. "Nipsic" on a three years' cruise in European waters. On his return to the United States he was again on duty in the Bureau of Steam Engineering, this time for two years. His next service was on the U. S. S. "Iroquois" on the South Pacific station. After returning from this station he was made a past assistant engineer and assigned to the University of Tennessee, where he remained four years as instructor of mechanical engineering. His next service was on board the coast survey steamer "Thetis," followed by duty in connection with the trial trip of the U. S. S. "Oregon." In 1897 he was made a chief engineer, and after a brief service in Mare Island Navy Yard crossed the Pacific Ocean as chief engineer of the monitor "Monadnock." In 1899 he was made a lieutenant commander, and after service on the "Oregon" and "New Orleans" returned home and was connected with engineering duty at the Navy Yard, New York, followed by duty as inspector of engineering material. In 1903 he was appointed fleet engineer of the Caribbean Squadron, and in 1904 was promoted to the rank of commander. His last service was as inspector of engineering material. On June 30th of this year he was retired as captain. He died July 29th, 1905.

He became a member of this Society in June, 1883.

REUBEN GILBERT COLLINS.

Mr. Collins was born in Troy, N. Y., March 27th, 1860, and moved to Norwich, Conn., when he was ten years old; he attended school at the Norwich Academy and gained his mechanical experience as an apprentice in his father's shop; the firm was known as J. P. Collins & Co., and they were builders of heavy machinery and water-wheels. He spent one year in Worcester, Mass., with the Holyoke Machine Company as a millwright. In 1887 he became connected with the Tamarack-Osceola Manufacturing Co., and was employed by them to design and build their rolling mills at Dollar Bay, Michigan. After the works were completed he became the general superintendent. After leaving their employ he became superintendent of the Wyandotte Mining Co. of Michigan and remained in that position for about two years. In 1900 he accepted a position with the Anaconda Copper Mining Co. as assistant chief engineer in the construction of their new reduction works at Anaconda, and after the completion of the works he became mechanical superintendent for the works. In July, 1904, he moved to Seattle, Wash., and was associated as partner in a firm known as Gerrick, Collins & Gerrick, contractors for structural steel, and was engaged in this business at the time of his death, which occurred in Oakland, California, on February 9th, 1905. He was on a business trip to San Francisco, and no particulars are known in regard to the manner of his death except that he was found beside a railroad track entering Oakland, consequently must either have fallen from the train or been run over by the train.

He had had a wide experience in the construction and operation of rolling mills and reduction works, and had occupied various positions of trust and responsibility.

He became a member of the Society at the New York meeting in 1894.

GEORGE EDWARD DIXON.

Mr. Dixon was born in the city of Leeds, England, March 20th, 1850. His father, Edward Dixon, was an army contractor, being the third generation of the family to follow that business. His early education was acquired at private schools, after which he attended and graduated from Bramham College in Yorkshire.

In early youth Mr. Dixon held a strong liking for the engineer-

ing profession, but this was no doubt tempered somewhat by his love for a soldier's life. After leaving college he entered the offices of Messrs. Varvill & Sons of York, where he remained several years and learned the business of an ironmonger, a business he thoroughly detested.

In 1878 he was commissioned second lieutenant in the army, but in 1880 resigned from the Queen's Service with the rank of first lieutenant, and in April, 1881, removed with his family to the States, going as far west as Parsons, Kansas, and shortly after to Cincinnati, where he identified himself with the heating business, remaining there until 1885, when he moved to Chicago. During a career in that city of twenty years he was associated with the leading heating and ventilating engineers of the time. In 1888 he entered into partnership with Mr. Caldwell of the Goulds & Caldwell Manufacturing Co., but severing his connection with Mr. Caldwell in 1893 to found the firm now known as George E. Dixon & Co.—a firm Mr. Dixon lived to see take its place among the first heating and ventilating engineering firms of this country, some of the largest plants in the United States being installed by his firm. Associated with Mr. Dixon in the firm were his two sons, who now carry on the business so firmly established by him.

One of the first hot-water heating apparatus ever installed in a public building was designed by Mr. Dixon and erected in what is known as the Fairbanks Building in Chicago. This plant attracted a great deal of attention at the time, as it was generally held by heating engineers in this country that hot-water apparatus was inapplicable to the heating of large buildings. The plant was a great success and is in operation to-day.

Mr. Dixon designed and patented several heating appliances. Among them the Eclipse back-pressure valve, and the Dixon radiator shield.

He also designed, though never patented, the corner radiator valve which is now in such common use. The standard table of dimensions of fire-box boilers was also compiled by him for his own use, but is now in universal use by boiler manufacturers.

Mr. Dixon's death occurred June 26th, 1904, when motoring with his wife near Chicago. Their car was struck by an electric train and they were both instantly killed. Among his business associates Mr. Dixon was held in the highest esteem for his business integrity and his great capability as a mechanical engineer.

He was one of the pioneers in the heating industry of this country and contributed largely toward the development of that apparently almost perfect apparatus, the modern steam plant as applied to large buildings.

He joined the Society at the Washington meeting in May, 1887.

CARL FANNING EICKS.

Mr. Eicks was born March 23rd, 1880, in New York city, where he attended the public schools. In 1902 he graduated from Stevens' Institute. His shop experience was with the Richmond Locomotive Works, the Bath Iron Works, and the Virginia Iron, Coke & Coal Company.

On leaving the Institute he went to Denver, Colo., to accept a position, but returned shortly to associate himself with the Jones Underfeed Stoker Co. as salesman in their New York office, and later was made one of the representatives of that company in their main office at Montreal, Canada.

Owing to ill health he returned to New York and accepted a position with the C. W. Hunt Co. of West New Brighton, Staten Island, N. Y. His health failing, he gave up this position and went to Liberty, N. Y. His death occurred at that place on September 20th, 1904.

He connected himself with this Society at the New York meeting in 1902.

ANDREW FLETCHER.

Mr. Fletcher was born in Scotland on the 14th of March, 1829. He served the regular apprenticeship as a machinist in the Archimedes Works (Henry R. Dunham & Co.) of this city. While an apprentice he was marked by Mr. Dunham as an especially energetic and skilful workman, so much so that when barely out of his time he erected steamboat engines for Mr. Dunham. He was also sent by Mr. Dunham to Cuba at a critical period to erect and run, in one of the large sugar plantations some distance from Havana, the entire sugar plant machinery and engine required for same. On his return he still continued in the employ of Mr. Dunham, attracting general attention by his work on the steamers of that date, the principal ones being the then well-known steamers "Alida" and "Armenia."

In 1853 Mr. William Harrison, doing business on West Street near the Dunham works, had a son, Joseph G. Harrison, just of

age, who had served his time as machinist in the Cunningham works near by. Desiring to put his son into business and having noted the skill and personality of Andrew Fletcher and his elder brother William, he approached them with the ultimate result that the firm of Fletcher, Harrison & Co. was formed, and they established what was then and is now known as the North River Iron Works. It so happened that just at this time Mr. Henry R. Dunham retired from business, so that there was no friction between the former employer of the Fletchers and the new concern. The North River Iron Works commenced business making repairs on river steamers, but it was not long before a contract was given to them to build an engine for the first steamer to run between New York and Harlem on the East River—the steamer "Sylvan Shore." This steamer was so successful that the following year the steamer "Sylvan Grove" was ordered and the engine was built by the North River Iron Works. From this time to 1860 the firm built a number of river steamers, every one of which was a distinct success, but Mr. Andrew Fletcher's opportunity came in the year 1861. At this time the fastest Hudson River steamer was the "Thomas Powell," commanded by Captain A. E. Anderson and owned by himself and two others. Desiring to build a better boat than the "Thomas Powell," Captain Anderson conferred with Mr. Andrew Fletcher, who agreed to build the machinery for a steamer to be one-half hour faster between New York and Rondout than the "Thomas Powell." The contract for the hull was given to Mr. Michael Allison of Jersey City, and for the machinery to Fletcher, Harrison & Co., the result being that in 1862 the famous steamer "Mary Powell" made her appearance on the river and proved herself to be then, and for many years after, the fastest river steamer of this country.

The reputation gained by Fletcher, Harrison & Co., and by Mr. Andrew Fletcher in particular, through the steamer "Mary Powell," was a sufficient foundation for the future success of the firm, and from that time to the present no concern has been more uniformly successful in all its undertakings than the North River Iron Works, known consecutively as Fletcher, Harrison & Co.; W. & A. Fletcher, after the retirement of Mr. Harrison in 1880; and W. & A. Fletcher Co., a corporation formed after the death of Mr. William Fletcher in 1883. Mr. Andrew Fletcher was president and treasurer of the W. & A. Fletcher Co. from the time of its incorporation until his death.

This is the brief story of Mr. Fletcher as an engineer. As such he was not excelled, his natural ability being enhanced by his study of the various problems involved, which resulted in the formation of a judgment often sought by steamboat people throughout the United States. For instance, Mr. Van Santvoord, who controlled the Hudson River Day Line to Albany—the boats of which have always been so well and favorably known—invariably consulted Andrew Fletcher and would only give his contracts for changes of or for new machinery to the North River Iron Works; and when the Fall River Line desired to make a departure from the then magnificent wooden-hulled steamers "Bristol" and "Providence" and to build the steel-hulled steamer "Pilgrim," Mr. Andrew Fletcher's advice was sought by the late George Peirce, who became so well known as the constructor of the great Fall River Line steamers. The intimacy and confidence then formed resulted in the contracts being given to the W. & A. Fletcher Co. for the succeeding steamers "Puritan," "Plymouth," and "Priscilla," the success of every one of which is well known.

Andrew Fletcher was a born engineer who instinctively knew what was right and safe. Added to this he possessed an integrity that only permitted work to be done in the best manner and all contracts were finished in this spirit regardless of the letter and of cost.

He connected himself with the Society at the New York meeting in 1889.

GEORGE WILLIAM FRANK.

Mr. Frank was born November 28th, 1861, at Warsaw, N. Y. After preliminary education in the Brooklyn Polytechnic Institute and the Poughkeepsie Military Academy, he took the civil engineering course at the Rensselaer Polytechnic Institute of Troy, N. Y. While a student there he was employed during vacation periods on engineering work with the C. B. & Q. R. R.

In the summer of 1881 he built an impounding dam at Corning, Iowa, and had entire and responsible charge of the work.

In 1884 he became a resident of Kearney, Neb., and engaged in business there. In 1891 he took the special course in architecture at the Boston School of Technology and afterwards drafted the plans for various schools and buildings in Kearney. He assisted in the management of the G. W. Frank Improvement Co., which

included the Kearney Canal Co., the Street Railroad Co., and the Kearney Electric Light Co.

In 1898 he came to New York as assistant engineer and architect for J. G. White & Co., designing buildings for power plants, electric railways and the like.

His last active work was with the Whiting Reduction Co. in charge of the Yadkin River Hydraulic Department.

His death occurred at Liberty, N. Y., on January 19th, 1905.

He was associated with the American Society of Civil Engineers, the American Institute of Electrical Engineers, the National Geographical Society, and became a member of this Society in 1900.

ROBISON C. GREER.

Mr. Greer was born at Newport, Ky., December 2nd, 1866. His death occurred December 16th, 1904.

As a boy he manifested a decided taste for mechanics and was apprenticed to the machine shops of the Lane & Bodley Co. of Cincinnati, O., where subsequently he was advanced to the drafting room.

From 1892 to 1893 he made a specialty of the design and construction of roasters for ores and of general blast furnace machinery.

In 1895 he became one of the experts in the manufacture of Columbia bicycles, designing tools and inspecting quality.

He connected himself with this Society at the Niagara Falls meeting in 1898.

DAVID J. MATLACK.

Mr. Matlack had a career that was remarkable in many ways. He was born in Philadelphia 72 years ago, and after a common-school education entered the I. P. Morris Co.'s shops as an apprentice in February, 1849. After his apprenticeship he remained with the company as a journeyman. In 1863 he was made foreman, and in 1875 a member of the firm. When the I. P. Morris Co. became a part of the William Cramp & Sons Ship and Engine Building Co., Mr. Matlack was retained in his position as foundry superintendent. At the time of his death he had worked for the same concern for 55 years, which in itself is certainly a remarkable fact; but when we consider the man's work and what he has done, we see that this is a much more remarkable record.

For many years the foundry with which he was connected was

noted for turning out the largest and most intricate work cast in the United States. Mr. Matlack was recognized as one of, if not the foremost, foundryman in this country, if not in the world. Many engineers in different parts of the United States and Canada have consulted him concerning difficult work which they had on hand.

Some idea of the work carried on under Mr. Matlack can be formed when we consider a statement made by Mr. E. D. Leavitt, engineer for the Calumet & Hecla Copper Co., to the effect that Mr. Matlack had made castings valued at over $5,000,000 for his company alone.

At the time of his death, July 25th, 1904, Mr. Matlack was apparently in perfect health, and the day he died had just superintended the pouring of a very large and difficult casting for a large turbine. He went to his suburban home as usual, but immediately after dinner died of heart disease.

Mr. Matlack was a member of the American Foundrymen's Association and of this Society, having joined at the New York meeting of 1883.

THOMAS R. MORGAN.

Mr. Morgan was born in South Wales in 1859, coming to this country with his father at the age of six.

He began work in his father's shop in Pittsburg while a mere boy. When not at school he spent all of his spare time in the shop, commencing at the bottom of the ladder. He was in turn foreman, engineer, machinist, and he became a very skilful draftsman.

He received a good common-school education and finished at Mount Union College at Alliance, O.

After leaving college he served a regular apprenticeship in the works at Alliance, to which place his father had removed from Pittsburg. The firm was first Marchand & Morgan, then Morgan, Williams & Co., and later The Morgan Engineering Co. He soon became superintendent, and at the age of 21 years was general manager, holding that position until he left the concern on the death of his father in 1897. To his ability, hard work and wonderful personality was due, next to his father, the success of that company.

He became connected with The American Society of Mechanical Engineers at the Pittsburg meeting in 1884.

In 1898 he became connected with The Wellman-Seaver-Engineering Company, the name of which was soon after changed to The Wellman-Seaver-Morgan Engineering Company, and later to The Wellman-Seaver-Morgan Company. He filled the position of Vice-President and works-manager of this company, and all associated with him can testify that he filled his position with ability. He was a good engineer, a splendid manager of men, and had the faculty of getting out of every man under him the best that was in him.

Like his friend and associate, Charles H. Wellman, he was beloved by all he came in contact with, and had hosts of friends both in business and private life.

He lost his life as the result of the railroad disaster of the 20th Century Limited at Mentor Station, Ohio, on the evening of July 21st, 1905.

He was in the smoking car of that train with his friends, Charles H. Wellman and Archibald P. Head, this being the car that was mostly demolished in the accident.

WILLIAM OFFUTT MUNDY.

Mr. Mundy was born at Louisville, Kentucky, receiving his technical education at the Rose Polytechnic Institute and graduating with the degrees of M.E. and E.E. in 1899.

His shop experience was one year as shop superintendent of the Louisville Railway Company, July, 1895–1896; three years as station superintendent of the Louisville Railway Company; and eight months with the Fred. M. Prescott Steam Pump Company of Milwaukee, Wisconsin. In March, 1900, he entered the engineering department of the General Electric Company at Schenectady, and from February, 1901, until May, 1902, was in charge of train control design. In May, 1902, he entered the employ of the St. Louis Transit Company as master mechanic of that company. August 1st, 1904, he entered the employ of the Westinghouse Electric Manufacturing Co. as commercial engineer in the railway and lighting departments.

Mr. Mundy joined the Society at its Saratoga Meeting, June, 1903. His death, March 29th, 1905, was caused by an abscess in the ear, developing into cerebro-meningitis.

EDWARD HARVEY PARKS.

Mr. Parks was born at St. Johnsbury, Vt., in 1837. He received his education at the public schools of Worcester.

In 1866 he entered the employ of the Brown & Sharpe Mfg. Co. as foreman of the machine shop, which position he held until 1879, when he became the mechanical engineer of the firm, having charge of the mechanical departments.

Some time after this he started in business for himself in Worcester, but in 1897 was again employed by Brown & Sharpe, assuming the position of designer in the engineering department.

His death occurred from heart disease on June 9th, 1905.

He joined the Society in 1881.

GEORGE H. PERKINS.

Mr. Perkins was born in the State of Illinois in 1839. He entered the tinsmiths' trade as an apprentice at fourteen years of age and became a shop foreman when still quite young.

In 1863, in New York, he entered into partnership with Mr. Joseph Le Comte, forming the firm of Le Comte & Perkins.

He took out many patents on machinery and processes for the manufacture of tin and other sheet-metal ware and for convenient handling of petroleum oil. He was also a specialist on the microscopic study of paraffin and various other petroleum products.

Among his most important patents, from a commercial standpoint, were machines for closing seams on tin cans, nailing boxes, and uniting tinned plates by means of heat and pressure without the use of flux or solder. His most important scientific inventions were an instrument for measuring viscosity and an apparatus for examining the physical condition of liquids at any given temperature.

He connected himself with the Society at the Atlantic City meeting in 1895, and his death occurred on February 2nd, 1905.

WILLIAM SELLERS.

Mr. Sellers was born in Upper Darby, Delaware County, Pa., September 19th, 1824. He was educated at a private school maintained by his father and two relatives for the education of their children, and served his apprenticeship at the machinists' trade with his uncle, John Morton Poole, of Wilmington, Delaware, for nearly seven years.

In 1845 he took charge of the large machine shop of Fairbanks, Bancroft & Co., in Providence, R. I. In 1848 he commenced the manufacture of machinists' tools and mill gearing in Philadelphia, and subsequently joined Edward Bancroft. The new firm was Bancroft & Sellers. Later John Sellers, Jr., was admitted as a partner, and in 1853 the new shop at 16th Street and Pennsylvania Avenue was occupied. Mr. Bancroft died about 1856, and the firm became William Sellers & Co. Finally, in 1886, the company was incorporated with William Sellers as president.

In 1868 he formed the Edgemoor Iron Co., of which he was president. This company furnished all of the iron structural material for the Centennial Exhibition buildings, and also all of the structural material for the Brooklyn Bridge, with the exception of the suspension cables.

In 1873 Mr. Sellers became president of the Midvale Steel Co., Nicetown, Philadelphia, Pa., which he reorganized and which under his management became the first successful producer of material required by the Government for its steel cannon.

In 1868 he became a member of the Board of Trustees of the University of Pennsylvania, and served continuously until the time of his death.

In 1864 Mr. Sellers was elected a member of the Philosophical Society, and in 1873 became a member of the Academy of Natural Sciences; he was a member of the Institute of Mechanical Engineers of Great Britain, the Iron and Steel Institute of Great Britain, a corresponding member of the Société d'Encouragement pour L'Industrie Nationale, in Paris, and at the close of the Paris Exposition, in 1899, the decoration of Chevalier de la Legion d'Honneur was conferred upon him. He was also a member of the American Society of Civil Engineers and of this Society, having been one of its founders in 1880.

Mr. Sellers was granted about ninety United States patents, the earliest one having been in 1857, and some were pending at the time of his death. A great variety of subjects were covered, including machine tools, injectors, rifling machines, riveters, boilers, hydraulic machinery, furnaces, hoists, cranes, steam hammers, steam engines, ordnance, turn tables, pumps, etc.

Probably the best known of his inventions is the spiral gear planer drive, in which the table is moved back and forth by a multi-thread screw on an inclined shaft engaging with a rack on the under surface of the table.

In 1847 Mr. Sellers was elected a member of the Franklin Institute. He served as a member of the Board of Managers from 1857 to 1861 and again from 1864 to 1892. He served as president from 1864 to 1867.

As chairman of a special committee of the Institute, he formulated a uniform system of screw threads which was presented to the Institute at a meeting on September 15th, 1864.

This report was approved by the Institute, and within a few years was adopted by the Government in its workshops by the leading railroad companies, prominent machine tool builders and others, under the various names of the United States, Sellers, or Franklin Institute Systems.

The leading difference between the Sellers and Whitworth forms lies in the angle of the thread and in the fact that it has a flat top and bottom instead of being round.

In 1890 the Navy Department at Washington sent out specifications to machine tool builders for an eight-foot turning and boring lathe for sixteen-inch steel cannon. The main bed was nine feet wide and over seventy-three feet long, the extension bed for the boring arrangement was about fifty-three feet long and five feet wide, making a total length of over one hundred and twenty-eight feet.

Mr. Sellers did not approve of the Government design and refused to bid, but he had a complete new design made, and convinced the Board of Engineers at Washington of the superiority of his design, which they adopted, discarding their own.

Mr. Sellers received the contract for this lathe, which weighed about 500,000 pounds.

In 1860, during a visit to England, his attention was called to the Gifford injector for feeding steam boilers. The device was crude, but Mr. Sellers saw in it the elements of a novel principle. He secured the American rights and commenced the manufacture of these injectors, which he slightly modified, however. In 1865, after various improvements, he invented and patented the self-adjusting combining tube which automatically adjusted the supply of water to meet the requirements of varying steam pressures. Further developments led to the most advanced and satisfactory forms of locomotive injectors.

Mr. Sellers died January 24th, 1905, in the 81st year of his age.

FREDERICK W. TAYLOR, JR.

Mr. Taylor was born at St. Johnsbury, Vt., December 28th, 1880. He received his education at the Union School and the St. Johnsbury Academy, graduating from the latter in 1899.

He spent the following year in the drafting department of E. & T. Fairbanks & Co. He then took a two-years' course at the Pratt Institute at Brooklyn, N. Y., graduating as a mechanical engineer in 1903.

He then returned to the Fairbanks Co. in the engineering department, which occupation he continued until the time of his death, which occurred very suddenly on July 24th, 1905.

He connected himself with the Society at the New York meeting in December, 1902.

WILLIAM G. VERNON.

Mr. Vernon was born in Delaware County, Pa., June 26th, 1844. He served an apprenticeship at the machinist trade, and then went with Pusey & Jones, Wilmington, Delaware, from which he later passed to the Baldwin Locomotive Works, where he served for fourteen years, of which six were as a contractor and eight as foreman of one of the shops.

Subsequently he connected himself with the firm of Goodell & Waters as partner, making a specialty of wood-working machinery. In the spring of 1895 he sold out his interest in the firm of Goodell & Waters and became one of the owners and general manager of the Philadelphia Lawn Mower Co. of this city.

He connected himself with the Society at the New York meeting in 1889, and his death, May 12th, 1904, resulted from apoplexy.

CHARLES H. WELLMAN.

Mr. Wellman was born June 12th, 1864, in Nashua, N. H. After the ordinary school training he went to Alliance, O., where he served two years, from 1884 to 1886, as machinist apprentice in the works of The Morgan Engineering Company, passing from the shop to the drafting-room of that company.

From 1886 to 1887 he had charge of the drawing-room of The Otis Steel Co., and was promoted from that position to superintendent of the new open hearth department, staying with that company until the latter part of 1889.

From 1890 to 1895 he was superintendent of The Wellman Iron and Steel Company at Thurlow, Pa.

He became a member of The American Society of Mechanical Engineers at the June meeting in Detroit in 1895.

From Thurlow he went to Chicago, and had charge, as superintendent, of the new Open Hearth steel plant and plate mill of the Illinois Steel Company at their South Chicago works, filling that position with signal ability from 1895 to 1896.

He left Chicago in 1896 to become associated with his brother, S. T. Wellman, and Mr. J. W. Seaver in the organization of The Wellman-Seaver Engineering Company at Cleveland, Ohio, as its general manager. The great success of that company in the line of iron and steel engineering and construction was very largely due to his ability and hard work.

He was an untiring worker and was of such a genial personality that he made friends with every one that he came in contact with, no matter whether it was the common workman in the steel works or the millionaire in his home or office. He was always pleasant and had a kind word for every one, and always looked on the bright side of everything.

His death took place June 22nd, 1905, as one of the consequences of the wreck of the 20th Century Limited at Mentor Station, Ohio, due to a misplaced switch on the line. He was on the train with his friend Mr. T. R. Morgan and Archibald P. Head of London, who were also victims of the same disaster.

CHARLES MASON WILKES.

Mr. Wilkes was born in Manchester, Conn., May 29th, 1858; received the degree of B.S. from Massachusetts Institute of Technology in 1881. His professional work since graduation has been in connection with the Massachusetts Drainage Commission in the interests of the city of Boston, and upon the sewage system of the city of St. Paul. He was assistant chief engineer of the World's Fair in 1893, having charge of the water, sewage and fire protection department, and was in charge of the construction in all of the above systems.

When he connected himself with the Society at the New York meeting of December, 1895, he had been engineer with the firm of D. H. Burnham & Company, making a specialty of sanitation, steam heating and ventilation of large buildings, and continued

with them until his death, designing all of the power, light, electric and sanitary plants erected by them.

He was engaged in this work at the time of his death, which took place at Philadelphia as the result of Bright's disease, January 7th, 1905.

HERMAN F. WITTE.

Mr. Witte was born July 16th, 1860, at Blue Island, Ill. He served his apprenticeship with the American Steam Engine Works of Chicago. In 1887 he became foreman of the Crane Elevator Company, and in 1889 became the general superintendent of the Otis Elevator Company at Yonkers, N. Y.

He was the inventor of special designs for an hydraulic valve for high pressure when the pilot valve is operated on low pressure for elevator service.

He connected himself with the Society at the New York meeting in 1902, and his death occurred at Yonkers on June 26th, 1904.

HORACE W. WYMAN.

Mr. Wyman was born in Worcester, May 30th, 1861. He graduated from the Worcester Free Institute in 1882, and then worked as a draftsman at the Crompton Loom Works.

Then the firm of Wyman & Gordon made its start in a small shop of two rooms, one for drop-forging, and one for the machine work, at 30 Bradley Street, where the present large buildings stand. Gradually, but steadily, the business has grown until 150 men are employed, and it is rated one of the most important drop-forging shops in the country.

In 1903 an additional plant was built in Cleveland, which has grown until it nearly equals the Worcester plant. Mr. Gordon removed to Cleveland and took charge of that shop on its completion in 1904. Beside his drop-forging business, Mr. Wyman, in 1889, in company with his father, Horace Wyman, organized the Worcester Storage Co., of which he was treasurer and manager. He was a trustee in the Worcester County institution for savings.

He connected himself with the Society at the Chicago meeting of May, 1886.

His death occurred early in October, 1905.

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